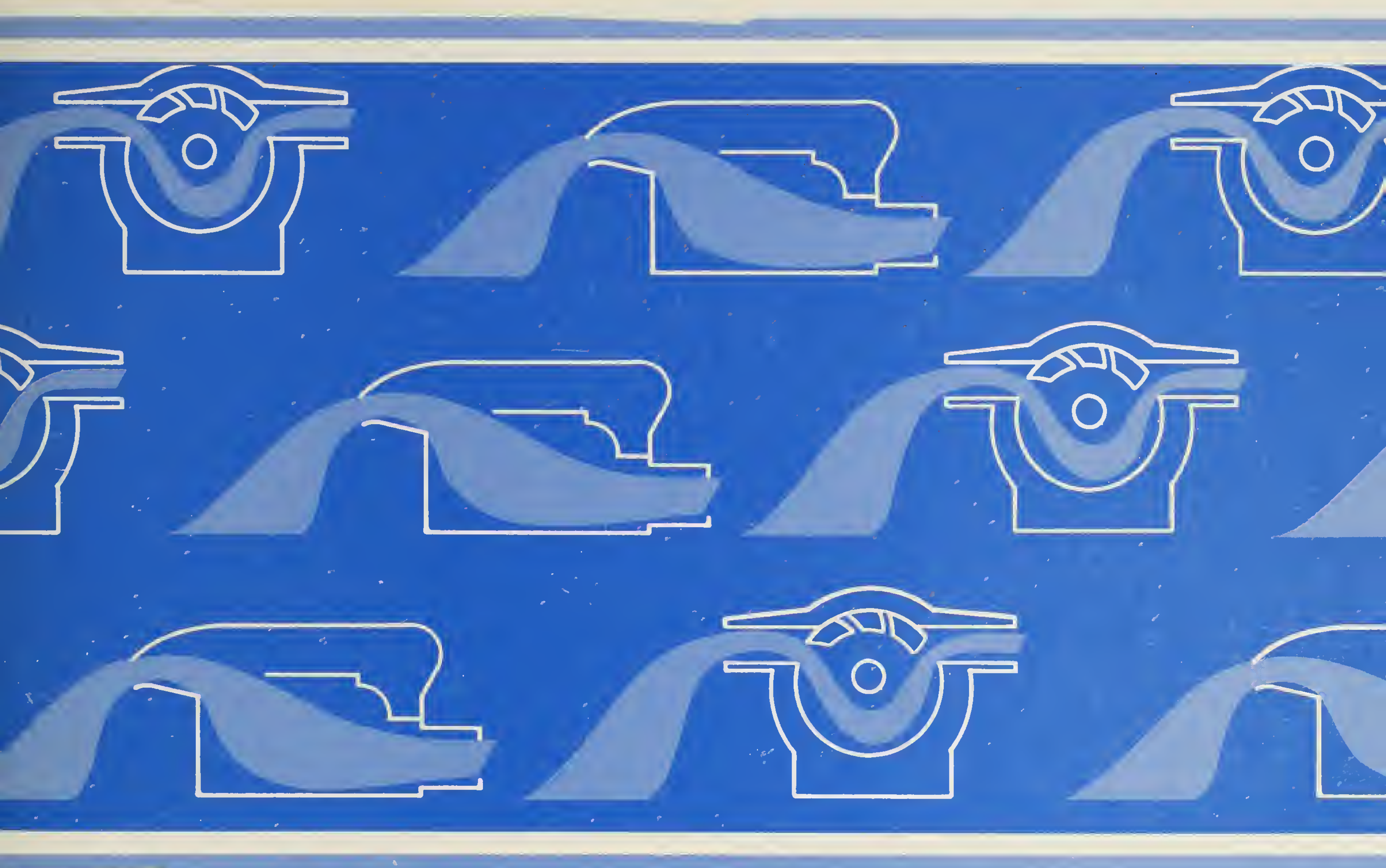
Agriculture Canada

Handling Agricultural Materials

Liquid conveyors



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FOREWORD

Handling Agricultural Materials is produced in several parts as a guide to designers of materials-handling systems for farm and associated industries. Sections deal with selection and design of specific types of equipment for materials handling and processing. Items may be required to function independently or as components of a system. The design of a complete system may require information from several sections of the manual.

This section was prepared by UMA Engineering Ltd., Winnipeg, Man., for the Canada Committee on Agricultural Engineering Services of the Canadian Agricultural Services Coordinating Committee.

1 INTRODUCTION

The term liquid applies to any substance that deforms under shear stress while the spacing between its molecules remains essentially constant. In other words, liquids are incompressible fluids. In contrast, gases demonstrate similar properties to liquids, yet do compress.

Liquid conveyors – or pumps – economically and conveniently transport liquids for all types of farm and industrial applications. To ensure efficient operation of liquid-conveying systems, however, the equipment and materials selected must match the type of liquid conveyed and the operational requirements.

1.1 Properties of liquids

Designing a liquid-handling system involves an understanding of both the behavior of the system and the properties of the material handled. Eight important properties of liquids stand out:

- specific weight
- mass density
- relative density
- viscosity
- kinematic viscosity
- vapor pressure
- specific heat capacity
- bulk modulus

In addition to these eight characteristics, also consider the corrosion and toxic properties of liquids when designing and selecting equipment for a liquid-handling system. Abrasion, too, becomes a consideration when the liquid contains suspended solids.

1.2 *Specific weight* Specific weight (γ) refers to the gravitational force, or weight, per unit volume of a given liquid. Specific weight varies with temperature. For instance, the specific weight of water at 4°C is 9810 N/m^3. The SI standard does not recognize the term specific weight; however, most sources continue to reference this property, measured in N/m^3.

1.3 *Mass density* The term mass density (ρ) describes the mass of liquid per unit volume and equals the specific weight at sea level divided by the gravitational constant (g). Mass density is often simply called density. The mass density of water at 4°C and at atmospheric pressure is 1000 kg/m^3.

1.4 *Relative density* The ratio of the mass density of a given liquid to the mass density of water at a standard reference temperature (generally 4°C) is known as relative density (d). Because relative density is a ratio of densities, it has no dimensions and is therefore independent of the units employed. The term relative density replaces the Imperial system term specific gravity.

Measure the relative density of liquids with a hydrometer. Hydrometers to suit various needs are readily available. The hydrometer scale is graduated to read relative density based on several references. Water is the most common; however, other reference scales exist to suit various applications. The American Petroleum Institute (API) scale is used to specify the relative density of petroleum products. The Baumé scale applies in the food industry. The Brix scale specifies the relative density of molasses, honey, and sugar syrups. API, Baumé, and Brix measurements are specified in degrees.

API gravity, a function of relative density, may be expressed as:

$$\text{API} = \frac{141.5}{d} - 131.5$$

For liquids heavier than water:

$$°\text{Baumé} = 145 - \frac{145}{d}$$

For liquids lighter than water:

$$°\text{Baumé} = \frac{140}{d} - 130$$

For the three preceding equations, the relative density is equal to the ratio of the density of the liquid to the density of distilled water, both densities measured at 42.2°C (Perry and Chilton: *Chemical Engineers' Handbook*).

°Brix = relative density of a 1% solution of sugar

1.5 *Viscosity* The property of a liquid that causes it to resist shear motion between adjacent particles is called viscosity (μ). This property could be considered the liquid's internal friction. The SI units of viscosity are pascal seconds (Pa·s), which are equal to kg/(m·s). Previously (in imperial units) the term poise, measured in g/(cm·s), described viscosity.

Viscosity in a liquid is proportional to the strain over time. This relationship implies that the faster a particle moves through the liquid the greater the resistance that particle encounters.

1.6 *Kinematic viscosity* The kinematic viscosity (ν) of a fluid is its viscosity divided by its density. The square metre per second (m^2/s) is the SI unit of kinematic viscosity. It supersedes the imperial unit, stoke (cm^2/s).

In practice absolute viscosity is awkward to measure directly. More commonly, scientists and engineers use indirect methods. For example, the measurement of the torque required to rotate a spindle at constant speed in the liquid establishes absolute viscosity. Rotating spindle viscosimeters, often called Brookfield viscosimeters, are available for most liquids.

Likewise, measuring the time for a specific volume of liquid to flow through an orifice or tube under its own head determines kinematic velocity. Numerous viscosimeters have been developed and procedures for determining viscosity have been standardized. Refer to ASTM designation D445-74 for one such procedure, which uses calibrated glass capillary instruments with gravity flow.

Alternatively, Saybolt viscosimeters (ASTM designation D88-56) measure flow rates under prescribed head and temperature conditions as the liquids flow through a short, small-bore tube. Viscosity is reported as the time (in seconds) for 60 mL of liquid to flow through the orifice. Use the Saybolt universal viscosimeter for light and medium liquids. Use the Saybolt Furol, with its larger bore, for heavy liquids.

Table 1 lists approximate conversions for several commonly used viscosity measurements including the SI system. Table 2 provides the viscosity of several common liquids. The publication ASTM D2146-74 provides a complete set of tables for converting kinematic viscosity to Saybolt Universal or Saybolt Furol viscosity.

1.7 *Vapor pressure* At the surface of every liquid, evaporation takes place until the pressure in the space above the surface prevents further molecular exchange. This pressure is called the saturated vapor pressure (p_v). Because the rate of evaporation depends on temperature, the vapor pressure is a function of the temperature of the liquid. Thus, either increasing temperature or decreasing pressure causes boiling. The vapor pressure of water varies from 0.61048 kPa at 0°C to 101.32 kPa at 100°C.

1.8 *Specific heat capacity* Specific heat capacity (c) describes the quantity of heat required to raise a unit mass of a substance one degree. This parameter measures the capacity of substances to store energy. The SI units of specific heat capacity are joules per kilogram-degree Kelvin, J/(kg·K).

1.9 *Bulk modulus* The bulk modulus (B_m) of elasticity of a liquid relates to the amount of deformation a given pressure change can cause. In other words:

$$B_m = \frac{(p_2 - p_1) \times v_1}{v_1 - v_2}$$

where p_1 = initial pressure
p_2 = final pressure
v_1 = initial volume
v_2 = final volume

1.10 Behavior of liquids

The behavior of liquids involves five properties:

- static pressure
- gage pressure
- absolute pressure
- energy at rest
- kinetic energy

1.11 *Static pressure* At every point in a liquid at rest, a pressure acts equally in all directions. This quantity is the static pressure, or simply pressure. This pressure varies only with change in elevation.

1.12 *Gage pressure* The difference between the pressure of a given liquid and that of the atmosphere is called gage pressure.

1.13 *Absolute pressure* Absolute pressure equals the gage pressure plus atmospheric pressure. For a homogeneous fluid exposed to atmospheric pressure the absolute pressure may be expressed as:

$$p_{abs} = p_a + H \times \rho \times g$$

where p_{abs} = absolute pressure (Pa)
p_a = atmospheric pressure (Pa)
ρ = mass density (kg/m^3)
H = head of liquid above point where pressure is measured (m)
g = gravitational constant = 9.81 m/s^2

Head, measured in units of height of the given liquid, is frequently referred to as static head.

1.14 *Energy at rest* The energy of a liquid at rest consists of the potential energy due to its elevation, the potential energy due to its gage pressure, and the elastic energy. In most agricultural applications, however, assume no elastic energy because pressures are relatively low and most of the liquids used are incompressible.

Table 1 Viscosity comparisons

Kinematic viscosity		Absolute viscosity (Pa·s)						
	Saybolt universal seconds	Relative density						
m^2/s	(SUS)	0.8	0.9	1.0	1.1	1.2	1.3	1.4
2.10×10^{-2}	10×10^4	16.8	18.9	21.0	23.1	25.2	27.3	29.4
1.89×10^{-2}	9×10^4	15.1	17.0	18.9	20.8	22.7	24.6	26.4
1.68×10^{-2}	8×10^4	13.4	15.1	16.8	18.5	20.2	21.8	23.5
1.47×10^{-2}	7×10^4	11.7	13.2	14.7	16.2	17.6	19.1	20.6
1.26×10^{-2}	6×10^4	10.1	11.3	12.6	13.9	15.1	16.5	17.6
1.05×10^{-2}	4.0×10^4	8.40	9.45	10.5	11.6	12.6	13.7	14.7
9.45×10^{-3}	4.5×10^4	7.56	8.50	9.45	10.4	11.4	12.3	13.2
8.50×10^{-3}	4.0×10^4	6.80	7.65	8.50	9.35	10.2	11.1	11.9
7.35×10^{-3}	3.5×10^4	5.88	6.62	7.35	8.09	8.83	9.56	10.3
6.30×10^{-3}	3.0×10^4	5.04	5.67	6.30	6.94	7.56	8.20	8.83
5.25×10^{-3}	2.5×10^4	4.20	4.72	5.25	5.78	6.30	6.83	7.35
4.25×10^{-3}	2.0×10^4	3.40	3.82	4.25	4.68	5.10	5.53	5.95
3.15×10^{-3}	1.5×10^4	2.52	2.84	3.15	3.46	3.78	4.09	4.41
2.20×10^{-3}	1.0×10^4	1.76	1.98	2.20	2.42	2.64	2.86	3.08
1.95×10^{-3}	9×10^3	1.56	1.75	1.95	2.15	2.34	2.53	2.73
1.70×10^{-3}	8×10^3	1.36	1.53	1.70	1.87	2.04	2.21	2.38
1.50×10^{-3}	7×10^3	1.20	1.35	1.50	1.65	1.80	1.95	2.10
1.30×10^{-3}	6×10^3	1.04	1.17	1.30	1.43	1.56	1.69	1.82
1.05×10^{-3}	5×10^3	0.840	0.945	1.05	1.15	1.26	1.37	1.47
8.50×10^{-4}	4×10^3	0.680	0.765	0.850	0.935	1.02	1.10	1.19
6.30×10^{-4}	3×10^3	0.505	0.567	0.630	0.694	0.756	0.820	0.883
4.20×10^{-4}	2×10^3	0.336	0.378	0.420	0.462	0.504	0.546	0.588
2.20×10^{-4}	1×10^3	0.176	0.198	0.220	0.242	0.264	0.286	0.308
1.95×10^{-4}	9×10^2	0.156	0.175	0.195	0.214	0.234	0.253	0.273
1.70×10^{-4}	8×10^2	0.136	0.153	0.170	0.187	0.204	0.221	0.238
1.50×10^{-4}	7×10^2	0.120	0.135	0.150	0.165	0.180	0.195	0.210
1.30×10^{-4}	6×10^2	0.104	0.117	0.130	0.143	0.156	0.169	0.182
1.05×10^{-4}	5×10^2	0.084	0.094	0.105	0.109	0.126	0.136	0.147
8.50×10^{-5}	4×10^2	0.086	0.077	0.085	0.094	0.102	0.111	0.119
6.30×10^{-5}	3×10^2	0.050	0.057	0.063	0.069	0.076	0.083	0.088
4.20×10^{-5}	2×10^2	0.034	0.038	0.042	0.046	0.050	0.055	0.059
2.20×10^{-5}	1×10^2	0.018	0.020	0.022	0.024	0.026	0.029	0.031
1.90×10^{-5}	90	0.015	0.017	0.019	0.021	0.023	0.025	0.027
1.70×10^{-5}	80	0.014	0.015	0.017	0.019	0.020	0.022	0.024
1.50×10^{-5}	70	0.012	0.014	0.015	0.017	0.018	0.020	0.021
1.0×10^{-5}	60	0.008	0.009	0.010	0.011	0.012	0.013	0.014
7.4×10^{-6}	50	0.006	0.006	0.007	0.008	0.008	0.009	0.010
4.2×10^{-6}	40	0.003	0.004	0.004	0.004	0.005	0.005	0.006
1.2×10^{-6}	30	0.001	0.001	0.001	0.001	0.002	0.002	0.002

Table 2 Relative density and viscosity of liquids

Material	Temperature (°C)	Relative density	Viscosity (Pa·s)
Water	0	1.000	1.79×10^{-3}
	20	0.998	1.00×10^{-3}
	50	0.987	5.49×10^{-4}
Sucrose			
20%	0	1.086	3.82×10^{-3}
	20	1.082	1.92×10^{-3}
	80	1.055	5.92×10^{-4}
60%	20	1.289	6.02×10^{-2}
	80	1.252	5.42×10^{-3}
Lube oil			
SAE 10	15	0.900	1.00×10^{-1}
	65	0.870	1.00×10^{-2}
SAE 30	15	0.900	4.00×10^{-1}
	65	0.870	2.70×10^{-2}
$CaCl_2$			
23%	0	1.220	3.60×10^{-3}
	−20	1.220	5.91×10^{-3}
29%	−30	1.280	1.08×10^{-2}
Molasses, heavy dark	20	1.400	6.60
	40	1.370	1.87
	50	1.13	9.20×10^{-1}
Soybean oil	30	0.92	4.06×10^{-2}
Olive oil	20	0.92	8.40×10^{-2}
Rapeseed	15	0.91	1.18×10^{-1}
Milk			
Whole	0	1.04	4.28×10^{-3}
	20	1.03	2.12×10^{-3}
Skim	25	1.04	1.37×10^{-3}
Cream			
20%	3	1.01	6.20×10^{-3}
30%	3	1.00	1.38×10^{-3}

1.15 *Kinetic energy* For a liquid in motion include the kinetic energy. Assuming no change in internal energy and no transfer of energy in or out of the system the total energy must remain constant, thus:

$$C = \frac{V^2}{2g} + H + \frac{p}{\gamma}$$

where C = constant
V = velocity
H = head
p = pressure
γ = specific weight

This equation assumes no transfer of energy between the liquid and its surroundings. In practice, though, the total energy contained by the liquid may change.

- Heat may transfer between the liquid and its surroundings.
- The flowing material may do work (called shaft work, W_s) on the surroundings and consequently may lose energy.
- Friction losses (F) occur between the flowing material and the surroundings.

The following equation, called Bernoulli's equation, expresses the change in energy between two points in a system. Assume no change in density and no heat transfer.

$$\frac{V_1^2}{2g} + H_1 + \frac{p_1^2}{\gamma} = \frac{V_2^2}{2g} + H_2 + \frac{p_2}{\gamma} + F + W_s$$

where V_1 = initial velocity
V_2 = final velocity
H_1 = initial head
H_2 = final head
p_1 = initial pressure
p_2 = final pressure
γ = specific weight
F = friction
W_s = shaft work

2 PROPERTIES OF PIPE AND TUBING

2.1 General characteristics

Select pipe materials on the basis of the requirements set by codes and standards established by such organizations as the Canadian Standards Association (CSA) or the American Society of Testing Materials (ASTM).

For most farm applications, a wide range of pipe materials is available to meet the appropriate standards. Ultimately, safety, durability, and cost guide the selection of pipe equipment.

2.2 Steel pipe and tubing

CSA 63 pipe, also designated ASTM A120, is considered the basic standard for steel pipe. It is available in either black or galvanized metal and in three classes of wall thickness. Table 3 lists physical dimensions.

Table 3 Steel pipe dimensions and schedule numbers

Nominal size (in.)	Outside diameter (in.)	Outside diameter (mm)	Wall thickness (in.)	Wall thickness (mm)	Weight* class	Schedule number
0.13	0.405	10.3	0.068	1.7	STD	40
			0.095	2.4	XS	80
0.25	0.540	13.7	0.088	2.2	STD	40
			0.119	3.0	XS	80
0.38	0.675	17.1	0.091	2.3	STD	40
			0.126	3.2	XS	80
0.50	0.840	21.3	0.109	2.8	STD	40
			0.147	3.7	XS	80
			0.188	4.8		160
			0.294	7.5	XXS	
0.75	1.050	26.7	0.113	2.9	STD	40
			0.154	3.9	XS	80
			0.219	5.6		160
			0.308	7.8	XXS	
1.00	1.315	33.4	0.133	3.4	STD	40
			0.179	4.5	XS	80
			0.250	6.4		160
			0.358	9.1	XXS	
1.25	1.660	42.2	0.140	3.6	STD	40
			0.191	4.9	XS	80
			0.250	6.4		160
			0.382	9.7	XXS	
1.50	1.900	48.3	0.145	3.7	STD	40
			0.200	5.1	XS	80
			0.281	7.1		160
			0.400	10.2	XXS	
2.00	2.375	60.3	0.154	3.9	STD	40
			0.218	5.5	XS	80
			0.344	8.7		160
			0.436	11.1	XXS	
2.50	2.875	73.0	0.203	5.2	STD	40
			0.276	7.0	XS	80
			0.375	9.5		160
			0.552	14.0	XXS	
3.00	3.500	88.9	0.216	5.5	STD	40
			0.300	7.6	XS	80
			0.438	11.1		160
			0.600	15.2	XXS	
3.50	4.000	101.6	0.226	5.7	STD	40
			0.318	8.1	XS	80
4.00	4.500	114.3	0.237	6.0	STD	40
			0.337	8.6	XS	80
			0.438	11.1		120
			0.531	13.5		160
			0.674	17.1	XXS	

(continued)

Table 3 **Steel pipe dimensions and schedule numbers** *(concluded)*

Nominal size (in.)	Outside diameter (in.)	(mm)	Wall thickness (in.)	(mm)	Weight* class	Schedule number
5.00	5.563	141.3	0.258	6.6	STD	40
			0.375	9.5	XS	80
			0.500	12.7		120
			0.625	15.9		160
			0.750	19.1	XXS	
6.00	6.625	168.3	0.280	7.1	STD	40
			0.432	11.0	XS	80
			0.562	14.3		120
			0.719	18.3		160
			0.864	21.9	XXS	
8.00	8.625	219.1	0.250	6.4		20
			0.277	7.0		30
			0.322	8.2	STD	40
			0.406	10.3		60
			0.500	12.7	XS	80
			0.594	15.1		100
			0.719	18.3		120
			0.812	20.6		140
			0.875	22.2	XXS	
			0.906	23.0		160
10.00	10.750	273.1	0.250	6.4		20
			0.307	7.8		30
			0.365	9.3	STD	40
			0.500	12.7	XS	60
			0.594	15.1		80
			0.719	18.3		100
			0.844	21.4		120
			1.000	25.4	XXS	140
			1.125	28.6		160
12.00	12.750	323.9	0.250	6.4		20
			0.330	8.4		30
			0.375	9.5	STD	
			0.406	10.3		40
			0.500	12.7	XS	
			0.562	14.3		60
			0.688	17.5		80
			0.844	21.4		100
			1.000	25.4	XXS	120
			1.125	28.6		140
			1.312	33.3		160

* STD, standard; XS, strong; XXS, double strength.

Source: *Canadian Standards Association CSA Z245.1-M1982.*

Use this basic pipe in steam, water, gas, and air lines, including plumbing, sprinklers, heating, and air conditioning applications. Do not use it, however, for flammable or toxic liquids. As well, avoid ASTM A120 pipe for close coiling or bending; use instead ASTM A53 pipe.

ASTM A120 pipe withstands a maximum pressure of 0.85 MPa and temperatures up to 205°C. Where tubing is subjected to high internal pressures, install steel boiler, condenser, heat exchanger, or hydraulic tubing. This kind of tubing is available in a wide variety of outside diameters (6–203 mm) and wall thicknesses (0.89–8.13 mm). Table 4 contains the data on available sizes from 3.18 to 50.80 mm.

Table 4 Hydraulic tubing pressure ratings (SI conversion from SAE J1065)

Nominal tube outside diameter (in.)	Nominal tube outside diameter (mm)	Nominal tube wall thickness (mm)	Reference working pressure (kPa)
0.13	3.18	0.71	38 600
		0.89	48 200
0.19	4.78	0.71	25 800
		0.89	32 000
0.25	6.35	0.71	19 300
		0.89	24 100
		1.24	33 800
		1.65	44 800
0.31	7.92	0.71	15 500
		0.89	19 300
		1.24	26 900
		1.65	35 800
0.38	9.52	0.71	12 700
		0.89	16 200
		1.24	22 400
		1.65	30 000
0.50	12.70	0.89	12 000
		1.24	16 900
		1.65	22 400
		2.11	28 600
0.63	15.88	0.89	9 600
		1.24	13 400
		1.65	17 900
		2.11	22 700
		2.41	26 200
0.75	19.05	0.89	7 900
		1.24	11 300
		1.65	14 800
		2.11	18 900
		2.41	21 700
		2.77	25 100
0.88	22.22	0.89	6 900
		1.24	9 600
		1.65	12 700
		2.11	16 200
		2.41	18 600
		2.77	21 300
1.00	25.40	0.89	6 000
		1.24	8 200
		1.65	11 000
		2.11	14 100
		2.41	16 200
		2.77	18 600
		3.05	20 700
1.13	28.58	1.24	7 600
		1.65	10 000
		2.11	12 700
		2.41	14 500
		2.77	16 500
		3.05	18 300

(continued)

Table 4 *(concluded)*

Nominal tube outside diameter (in.)	Nominal tube outside diameter (mm)	Nominal tube wall thickness (mm)	Reference working pressure (kPa)
1.25	31.75	1.24	6 900
		1.65	8 900
		2.11	11 400
		2.41	13 100
		2.77	15 100
		3.05	16 500
1.50	38.10	1.65	7 600
		2.11	9 600
		2.41	11 000
		2.77	12 406
		3.05	13 800
1.75	44.45	1.65	6 400
		2.11	8 300
		2.41	9 300
		2.77	10 700
		3.05	11 700
2.00	50.8	1.65	5 500
		2.11	7 200
		2.41	8 300
		2.77	9 300
		3.05	10 300
		3.40	11 400

2.3 Copper tubing

Copper tubing, suitable for plumbing and for conveying fluids, is available in three series of wall thicknesses designated K, L, and M (Table 5). Series K pipes are heavy-duty tubing useful for pump suction lines and underground piping. Series L tubing is standard weight suitable for inside buildings.

Both K and L types of tubing are available in annealed (soft) and hard-drawn (hard) forms. Use types K and L hard-tempered pipe as exposed piping inside buildings. It needs little support compared with soft-tempered tubing. For underground piping, use heavier type K tubing.

Series M tubing is light weight, suitable only for use behind walls inside buildings. Series M pipe is available in hard-tempered form, but it still requires physical protection.

Use the following expression to calculate the maximum working pressure for copper pipe.

$$p_w = 2\,S_t\,t\,/\,D_i F_s$$

where S_t = tensile strength (MPa)
= 207 MPa for copper
t = pipe wall thickness (mm)
D_i = inside diameter (mm)
F_s = safety factor, commonly 6

Table 5 Dimensions, weights, and diameter for standard copper water tubing

Standard size (mm)	Nominal outside diameter (mm)	Nominal wall thickness (mm)			Theoretical weight (kg/m)		
		type K	type L	type M	type K	type L	type M
6.4	9.525	0.889	0.762	*	0.22	0.19	*
9.5	12.700	1.245	0.889	0.635	0.40	0.29	0.22
12.7	15.875	1.245	1.016	0.711	0.51	0.42	0.30
15.9	19.050	1.245	1.067	*	0.62	0.54	*
19.0	22.225	1.651	1.143	0.813	0.95	0.68	0.49
25.4	28.575	1.651	1.270	0.889	1.25	0.97	0.69
31.8	34.925	1.651	1.397	1.067	1.55	1.32	1.01
38.1	41.275	1.829	1.524	1.245	2.02	1.70	1.40
50.4	53.975	2.108	1.778	1.473	3.06	2.60	2.17
63.5	66.675	2.413	2.032	1.651	4.36	3.69	3.02
76.2	79.375	2.769	2.286	1.829	5.95	4.96	3.99
88.9	92.075	3.048	2.540	2.108	7.62	6.38	5.33
101.6	104.775	3.404	2.794	2.413	9.69	8.01	6.93
127.0	130.175	4.064	3.175	2.769	14.39	11.32	9.91
152.4	155.575	4.877	3.556	3.099	20.54	15.2	13.27

* The material is not generally available.

Source: *ASTM designation B88-81 (SI conversion).*

2.4 Aluminum tubing

The document ASAE S263.2, published in the *American Society of Agricultural Engineers Year Book*, contains data on the minimum standards for aluminum irrigation tubing. In particular, the data prescribe the minimum requirements for tubing with outside diameters of 51, 76, 102, 127, 152, 178, 203, 229, 254, 279, 305, and 356 mm.

Because of the interrelationship between the wall thickness and diameter, and the mechanical properties of aluminum, system designs involving aluminum tubing generally do not prescribe a wall thickness for the pipe. Instead, the design must

- prevent denting
- protect from deflection or buckling
- withstand excessive torque loads

To prevent denting during handling, specify tubing with a denting factor equal or superior to the factors listed in Table 6. Calculate the denting factor (F_d) with this equation:

$$F_d = S_y t^2 / 1000$$

where S_y = specified yield strength of the material (MPa) (see Table 7)

t = wall thickness (mm)

When filled with water at zero pressure, the tubing must withstand a span 9.1 m as a simple beam without permanent deflection or local buckling. Any bending stress (σ_b) of the aluminum tubing that results cannot exceed the lesser of the two values found as follows:

σ_b = 90% of the specified yield strength

or

$$\sigma_b = 1.57 S_y - \frac{1.7\, S_y^2}{68\,947} \times \frac{D_o}{t}$$

where σ_b = bending stress

S_y = specified yield strength (MPa)

D_o = outside diameter (mm)

t = wall thickness (mm)

The applied bending stress

$$\sigma_b = M / \sigma_w$$

where $M = W L^2 / 8$ (maximum moment)

W = total load per metre of length (kN/m)

L = length of tubing (m)

$$\sigma_w = \frac{2\pi(D_o^4 - D_i^4)}{32 D_o}$$

D_o = outside diameter (mm)

D_i = inside diameter (mm)

σ_w = allowable working stress (see Table 7)

Table 6 Denting factors

Outside diameter (mm)	Denting factor (kN)
51.0	0.27
76.0	0.27
102.0	0.27
127.0	0.30
152.0	0.37
178.0	0.47
203.0	0.59
229.0	0.75
254.0	0.92

Denting factor = $S_y(t)^2/1000$.

Table 7 Minimum yield and maximum working stress for aluminum alloy tubing at room temperature

Alloy designation strength CSA HA series	Yield strength (MPa)	Maximum allowable working stress (MPa)	
		Non-welded	Arc-welded
3003-H112	34.5	21.0	21.0
6063-T6C	151.7	61.0	39.0
6061-T4	110.3	59.0	55.0
6061-T6	241.3	86.0	55.0
6351-T4A	144.8	80.0	55.0
6351-T6	255.1	96.0	55.0
5083-H111	165.5	91.0	78.0
5083-H321	213.7	100.0	78.0

Source: *Canadian Standards Association CSA169-M1978.*

For systems relying on mechanical devices to move pipes from location to location (e.g. irrigation systems), use a safety factor of 2 in designing devices to protect the tubing from excessive torque loads. Calculate the torque resistance (T) of aluminum tubing from the following equation:

$$T = 79\,290\,K \cdot D_0^{0.5} \cdot t^{2.5}$$

where K = stiffening factor

= 1 for extruded or welded tubing

D_0 = outside diameter (mm)

t = wall thickness (mm)

The modulus of elasticity for aluminum is 79 290 N/mm^2.

Determine the theoretical bursting pressure of tubing from the formula

$$p = 2\sigma_w \times t \,/\, D_0$$

where p = bursting pressure (MPa)

t = wall thickness (mm)

D_0 = outside diameter (mm)

The operating pressure of irrigation tubing should not exceed 1.034 MPa. On the basis of this maximum operating pressure, tubing that meets the ASAE Standard must withstand an internal hydrostatic pressure of 3.102 MPa for 2 min without leaking.

2.5 Plastic pipe and tubing

Most plastic pipe for domestic or agricultural use is manufactured from one of these materials:

- acrylonitrile–butadiene–styrene
- polybutylene
- polyethylene
- polyvinyl chloride
- chlorinated polyvinyl chloride

Acrylonitrile–butadiene–styrene (ABS) is a semirigid pipe that has pressure ratings between 550 and 1100 kPa. It is suitable for sewer pipe.

Polybutylene (PB) is a piping material useful for hot or cold potable water lines. PB pipe is rated at 1030 kPa. Check manufacturers' data for maximum stress and temperature before installing PB pipe. In addition, check local plumbing codes for acceptability.

Polyethylene (PE) is a flexible or semirigid pipe generally used for cold-water piping since its strength decreases as temperature rises. PE pipe is, however, useful for hot-water heating in concrete floors where temperatures up to 38°C are common. This kind of piping has pressure ratings between 550 and 1100 kPa.

Polyvinyl chloride (PVC) is a rigid pipe with pressure ratings of 340–2170 kPa. Use PVC for cold water only in such applications as household cold water pipes and drains, or permanent irrigation installations. For water pressure systems, the pipe should be rated at a minimum of 550 kPa. PVC pipe joins with couplings and solvent.

Chlorinated polyvinyl chloride (CPVC) is material similar to PVC but better suited to handling liquids at temperatures 22–33°C above the limit of other vinyl plastics. Use CPVC for hot or cold water lines.

In assessing material for use in the manufacture of plastic pipe, refer to the following CSA standards:

- CSA Standard B181.1-1973 acrylonitrile–butadiene–styrene drain, waste, and vent (ABS-DWV) pipe and pipe fittings
- CSA B137.8-M1977 polybutylene (PB) piping for hot and cold water distribution systems
- CSA B137.1-M1983 polyethylene pipe, tubing, and fittings for cold water pressure services
- CSA B137.3-M1981 rigid polyvinyl chloride (PVC) pipe for pressure applications
- CSA B137.6-M1983 CPVC pipe, tubing, and fittings for hot and cold water distribution systems

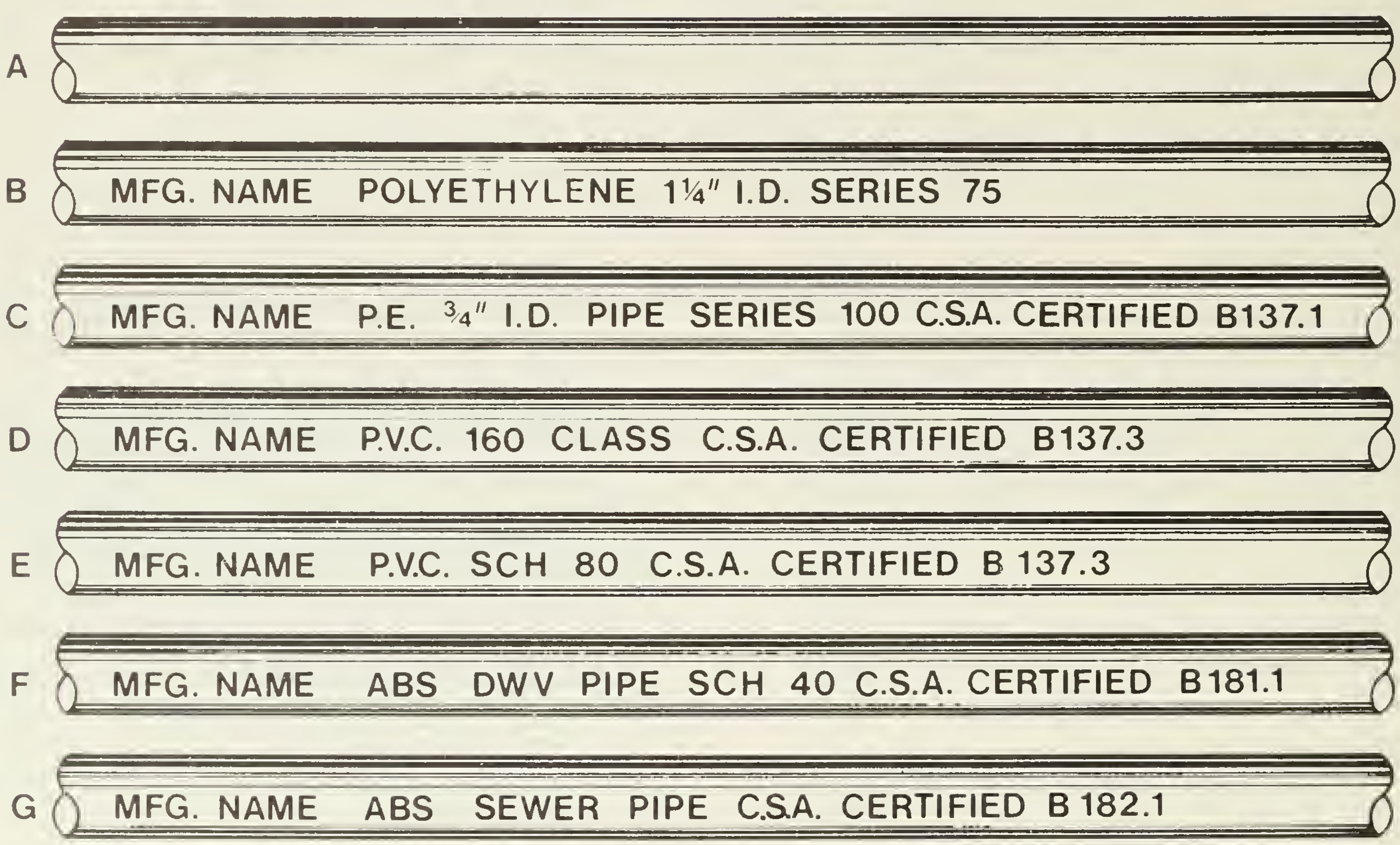
2.6 Using plastic pipe in pressure applications

Look for the following information on piping intended for use in pressure applications:

- material abbreviation
- type and grade, as defined in the appropriate CSA standard
- recommended hydrostatic design stress (RHDS) for water at 23°C

For example PVC 1220 means PVC Type 1, Grade 2, as defined in ASTM Standard D 1784 with an RHDS of 13.8 MPa (2000 psi). Fig. 1 illustrates pipe identification codes.

In applications where pipes contain liquids under pressure, use only pressure-rated plastic tubing. The pressure rating represents the estimated maximum internal pressure the pipe



A No coding. This is a utility pipe for use in residential and farm use. It may not be manufactured to CSA standard and should not be used for high pressure applications and it is recommended that it not be buried or installed in non-accessible locations.

B This is a polyethyline pipe manufactured to meet Series 75 CSA pipe dimensions but is not CSA certified.

C A polyethyline pipe which meets CSA Series 100 dimensional and pressure requirements.

D A polyvinyl chloride pipe which meets the pressure and dimensional requirements for CSA class 160 pipe.

E A PVC (rigid) pipe which meets the pressure and dimensional requirements of CSA schedule 40 pipe.

F Schedule 40 ABS pipe for use as a drain, waste or ventilation pipe. This pipe is not pressure rated.

G This pipe is manufactured from ABS material and is suitable for sewer applications. A perforated version may be used for sewage disposal fields.

Fig. 1. Plastic pipe identification coding.
Source: Agriculture Canada.

can withstand without failing. The pipe must be strong enough to handle both static and surge pressures.

Manufacturers classify pipe according to these pressure rating systems:

- schedule system
- class, series, or standard dimensional ratio system

2.7 *Schedule system* This system designates plastic pipe as schedule 40, 80, and 120. This designation is similar to that used for iron pipe where the outside diameter and wall thickness are fixed by specification. Each pipe size within each schedule has a recommended working pressure that depends on the dimensions of the pipe and the design stress of the pipe material. Generally, pipe used for industrial applications is classified under the schedule system.

Tables 8 and 9 illustrate pressure ratings for PVC pipe classified by the schedule system.

2.8 *Class, series, or standard dimensional ratio system* In this system plastic pipe is classified according to the working pressure. Each pipe in a single class or series has the same pressure rating. The pipe is grouped according to its standard dimensional ratio (SDR).

$$\mathrm{SDR} = \frac{\text{outside diameter}}{\text{wall thickness}}$$

Pipe that is pressure rated under the SDR system is classified as series or class *XXX*, where *XXX* is a two- or three-digit number representing the pressure rating in pounds per square inch for water at 23°C. At temperatures above 23°C plastic pipe loses fiber stress and can no longer withstand very high pressures. Table 10 illustrates how working pressure changes with temperature.

Common series or class numbers for plastic pipe are 50, 75, 100, 125, 160, and 200. The class or series to which a particular pipe is assigned depends on the pipe material and wall thickness. As a result, pipes with the same standard dimensional ratio but made from different materials do not necessarily fall in the same series or class.

Consider, for example, 50.8 mm of PVC pipe, type 1 series 160:

$$\begin{aligned} D_0 &= 60.325 \text{ mm} \\ t &= 2.311 \text{ mm} \\ \mathrm{SDR} &= 60.325/2.311 \\ &= 26 \\ p_m &= 160 \text{ psi} \\ &= 1100 \text{ kPa} \end{aligned}$$

where D_0 = outside diameter

t = wall thickness

p_m = maximum working pressure

See Table 11 for more examples.

Table 8 Minimum and maximum wall thicknesses for schedule pipe

Nominal diameter (in.)	Wall thickness (mm)			
	Schedule 40		Schedule 80	
	Minimum	Maximum	Minimum	Maximum
0.12	–	–	2.40	2.92
0.25	2.24	2.74	3.02	3.54
0.38	2.30	2.82	3.20	3.70
0.50	2.76	3.28	3.72	4.24
0.75	2.86	3.38	3.90	4.42
1.00	3.38	3.88	4.54	5.08
1.25	3.56	4.06	4.84	5.44
1.50	3.68	4.20	5.08	5.68
2.00	3.90	4.42	5.54	6.20
2.50	5.16	5.76	7.00	7.84
3.00	5.48	6.14	7.62	8.54
3.50	5.74	6.42	8.08	9.04
4.00	6.02	6.72	8.56	9.58
5.00	6.54	7.34	9.52	10.66
6.00	7.10	7.98	10.96	12.30
8.00	8.18	9.16	12.70	14.22
10.00	9.26	10.38	15.06	16.86
12.00	10.30	11.56	17.44	19.54

Source: *Canadian Standards Association CSA B137.3-M1981.*

Table 9 Pressure ratings for PVC schedule pipe

Nominal diameter (mm)	Pressure rating (kPa) PVC 12454	
	Schedule 40	Schedule 80
3.2	5580	8480
6.4	5380	7800
9.5	4270	6340
12.7	4140	5860
19.1	3300	4760
25.0	3100	4340
32.0	2550	3590
38.0	2280	3240
50.0	1930	2760
65.0	2070	2900
75.0	1790	2550
90.0	1650	2410
100.0	1520	2210
125.0	1310	2000
150.0	1240	1930
200.0	1100	1720
250.0	970	1590
300.0	900	1590

Source: *Canadian Standards Association CSA B137.3-M1981.*

Table 10 Reduction in working pressure for plastic pipes conveying material above 23°C

Temperature (°C)	Percent of stated working pressure at 23°C
23	100
25	95
30	83
35	72
40	60
45	47
50	36
55	24
60	13

Table 11 Standard dimensional ratio pipe pressure ratings

Dimensional ratio	Pressure rating (kPa) PVC 12454
41.0	690
32.5	860
26.0	1100
21.0	1380
17.0	1720

Source: *Canadian Standards Association CSA. B137.3-M1981*

2.9 General uses for plastic pipe

Use pipe and tubing that is not pressure rated as electrical conduit, or sewer, drain, and vent pipe. Some pipe manufacturers produce a utility pipe. Although this pipe is not CSA approved, it does suit minor drainage applications where the pipe is installed above ground. For underground installations or other major applications, use only CSA-approved pipe with the proper pressure rating. Table 12 lists some common applications with the pipe types suggested for each.

2.10 Plastic pipe fittings

Pipe manufacturers and distributors supply many standard pipe fittings, including:

- standard threaded fittings using the same size and thread system as schedule pipe
- insert fittings with clamps
- socket fittings, whereby the pipe slides into a socket and is cemented in place
- field-welded fittings in which a hot, inert gas welds the thermoplastic pipe at the joints
- patent joints

Table 12 Plastic pipe applications

Application	Pipe type	Remarks
Cold water service	PE (flexible) PVC (rigid) CPVC	Pipe should be pressure rated
Hot water	PB 21 PVC CPVC	Pipe should be pressure rated
Pump lines	PE (flexible) PVC (rigid)	 Class 160 or 200 or schedule 40 When used with a submersible pump a torque arrestor is recommended.
Sewer lines	PE (flexible) PVC (rigid)	 Solid or perforated pipe available.
Drain or vent pipe	ABS DWV pipe	
Underground drainage lines	PE (flexible and corrugated) PVC (rigid) ABS (rigid)	Solid or perforated Solid or perforated Solid or perforated
Conduit	PVC (rigid) Schedule 40 (rigid)	UL approved Not pressure rated
Chemical, petroleum products, etc.	Consult pipe manufacturer for recommendation.	

With patent joints, a bell with a seal is formed on one end of the pipe during manufacture. The connecting pipe end, usually beveled, inserts into the bell end. This system allows joint flexibility and rapid assembly at any temperature.

3 FLUID FRICTION IN PIPING SYSTEMS

When liquids flow in a conduit, frictional resistance occurs. The following Darcy–Weisbach equation estimates the energy loss due to friction.

$$H_f = \frac{F \cdot L \cdot V^2}{D \cdot 2g}$$

where H_f = head loss (m)

F = frictional resistance

L = length of pipe (m)

V = liquid velocity (m/s)

D = pipe diameter (m)

g = gravitational constant

= $9.81\ m/s^2$

The frictional resistance (F) in this equation is a function of the Reynolds number (Re) and of pipe roughness. Moody related these factors in a resistance diagram shown in Fig. 2. Table 18 lists values for the roughness factors for various types of pipe material.

The frictional resistance can also be calculated using these equations:

$$F = 64 / Re$$

For laminar flow where $Re < 2000$

$$F = 0.316 / Re$$

For turbulent flow in smooth pipes and where $3000 < Re < 100\,000$

$$Re = V{\cdot}D / v$$

where v = kinematic viscosity

When a piping system includes fittings, use Tables 13–17 to determine the equivalent length of pipe for each fitting. Add the equivalent lengths to the pipe lengths to establish the total length of pipe in the system. Then express the friction head loss for the system in metres per 100 m of total piping.

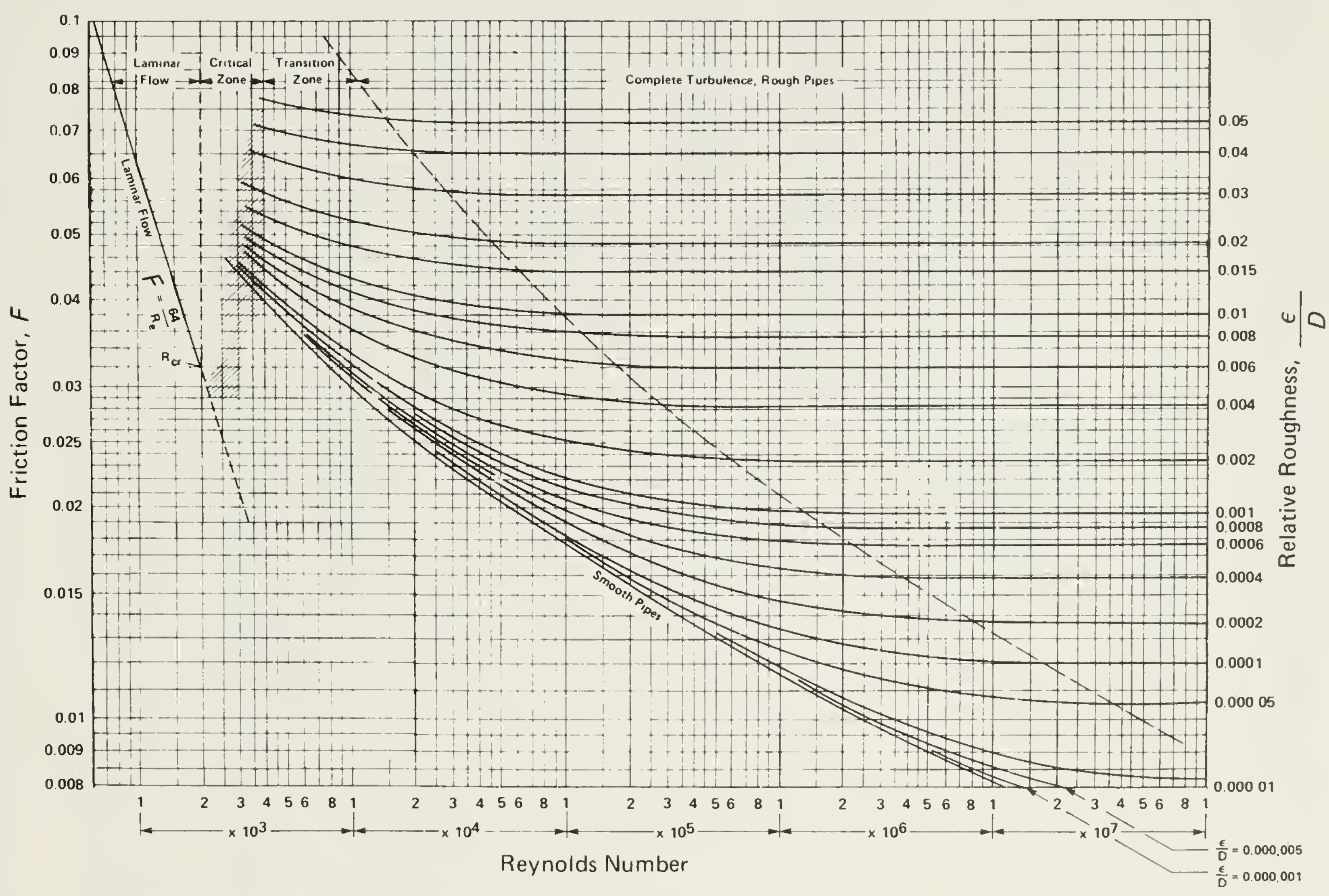


Fig. 2. Friction factors for fluid flow in pipes.
Source: **Agriculture Canada.**

Table 13 Friction head loss of water in schedule 40 steel pipe
(based on $C = 100$ in Hazen and Williams formula)

Flow rate (L/s)	Nominal pipe size (mm)										
	12	19	25	32	38	50	65	75	100	125	150
	m of water/100 m of pipe*										
0.1	4.9	1.2									
0.2	17.7	4.5	1.4								
0.3	37.5	9.5	2.9	0.8							
0.4	63.9	16.2	5.0	1.3	0.6						
0.5	96.5	24.5	7.6	2.0	0.9						
0.6	135.3	34.4	10.6	2.8	1.3						
0.7		45.8	14.1	3.7	1.7	0.5					
0.8		58.6	18.1	4.7	2.2	0.7					
0.9		72.9	22.5	5.9	2.8	0.8					
1.0		88.6	27.3	7.2	3.4	1.0					
1.5			57.9	15.2	7.2	2.1	0.9				
2.0			98.7	25.9	12.2	3.6	1.5	0.5			
2.5			149	39.2	18.5	5.5	2.3	0.8			
3.0				55.0	25.9	7.7	3.2	1.1			
3.5				73.1	34.5	10.2	4.3	1.5			
4.0				93.7	44.2	13.1	5.5	1.9	0.5		
4.5				116	55.0	16.3	6.8	2.4	0.6		
5.0				142	66.8	19.8	8.3	2.9	0.8		
5.5				169	79.7	23.6	9.9	3.4	0.9		
6.0					93.7	27.7	11.7	4.0	1.1		
6.5					109	32.2	13.5	4.7	1.2		
7.0					125	36.9	15.5	5.4	1.4		
7.5					142	41.9	17.6	6.1	1.6	0.5	
8.0					160	47.2	19.9	6.9	1.8	0.6	
8.5					178	52.9	22.2	7.7	2.1	0.7	
9.0						58.8	24.7	8.6	2.3	0.8	
9.5						64.9	27.3	9.5	2.5	0.9	
10						71.2	30.1	10.4	2.8	1.0	
12						100	42.1	14.6	3.9	1.3	0.5
14						133	56.0	19.5	5.2	1.7	0.7
16						170	71.8	24.9	6.6	2.2	0.9
18							89.3	31.0	8.2	2.7	1.1
20							108	37.7	10.0	3.3	1.4
25							164	56.9	15.2	5.0	2.1
30								79.8	21.2	7.1	2.9
35								106	28.3	9.4	3.8
40								136	36.2	12.0	4.9
45								169	45.0	15.0	6.1
50									54.7	18.2	7.4
55									65.3	21.7	8.9
60									76.7	25.5	10.4
65									88.9	29.6	12.1
70									102	33.9	13.9
75									116	38.6	15.8
80									131	43.4	17.8
85									146	48.6	19.9
90									162	54.0	22.1
95									180	59.7	24.4
100										65.7	26.8

* Head loss in m × 9.806 = head loss in kPa.

Table 14 Friction head loss of water in type L copper tubing
(based on $C = 130$ in Hazen and Williams formula)

Flow rate (L/s)	Nominal pipe size (mm)										
	13	16	19	25	32	38	50	65	75	100	150
	m of water/100 m of pipe*										
0.1	5.7	2.2	1.0								
0.2	20.7	7.8	3.5	0.9							
0.3	43.9	16.5	7.4	2.0	0.7						
0.4	74.8	28.2	12.6	3.4	1.2	0.5					
0.5	113	42.6	19.1	5.2	1.9	0.8					
0.6	158	59.7	26.8	7.3	2.6	1.1					
0.7		79.4	35.6	9.7	3.5	1.5					
0.8		102	45.6	12.4	4.5	1.9					
0.9		126	56.8	15.5	5.6	2.4	0.6				
1.0		154	69.0	18.8	6.7	2.9	0.7				
1.5			146	39.9	14.3	6.1	1.6	0.5			
2.0				67.9	24.4	10.5	2.7	0.9			
2.5				103	36.9	15.8	4.1	1.4	0.6		
3.0				144	51.7	22.2	5.8	2.0	0.8		
3.5					68.7	29.5	7.7	2.7	1.1		
4.0					88.0	37.8	9.8	3.4	1.4		
4.5					109	47.0	12.2	4.2	1.8		
5.0					133	57.1	14.8	5.2	2.2	0.5	
5.5					159	68.1	17.7	6.2	2.6	0.6	
6.0						80.0	20.8	7.2	3.0	0.8	
6.5						92.8	24.1	8.4	3.5	0.9	
7.0						106	27.6	9.6	4.0	1.0	
7.5						121	31.4	10.9	4.6	1.2	
8.0						136	35.4	12.3	5.2	1.3	
8.5						152	39.6	13.8	5.8	1.5	
9.0						169	44.0	15.3	6.4	1.6	
9.5							48.7	16.9	7.1	1.8	
10							53.5	18.6	7.8	2.0	
12							75.0	26.1	11.0	2.8	
14							100	34.7	14.6	3.7	0.5
16							128	44.5	18.7	4.7	0.7
18							159	55.3	23.3	5.9	0.8
20								67.3	28.3	7.1	1.0
25								102	42.7	10.8	1.5
30								142	59.9	15.2	2.1
35									79.7	20.2	2.8
40									102	25.8	3.6
45									127	32.1	4.5
50									154	39.0	5.5
55										46.6	6.5
60										54.6	7.7
65										63.5	8.9
70										72.8	10.2
75										82.7	11.6
80										93.2	13.1
85										104	14.6
90										116	16.2
95										128	18.0
100										141	19.8

* Head loss in m × 9.806 = head loss in kPa.

Table 15 Friction head loss of water in polyethylene pipe
(based on $C = 130$ in Hazen and Williams formula)

Flow rate (L/s)	Nominal pipe size (mm)								
	13	19	25	32	38	50	65	75	100
	m of water/100 m of pipe*								
0.1	2.6	0.7							
0.2	9.5	2.4	0.7						
0.3	20.1	5.1	1.6						
0.4	34.2	8.7	2.7	0.7					
0.5	51.8	13.2	4.1	1.1	0.5				
0.6	72.6	18.4	5.7	1.5	0.7				
0.7	96.5	24.5	7.6	2.0	0.9				
0.8	124.0	31.4	9.7	2.5	1.2				
0.9	154.0	39.1	12.1	3.2	1.5				
1.0		47.5	14.7	3.8	1.8	0.5			
1.5		101.0	31.0	8.2	3.8	1.1			
2.0		171.0	52.9	13.9	6.6	1.9	0.8		
2.5			80.0	21.0	9.9	2.9	1.3		
3.0			112.0	29.5	13.9	4.1	1.8	0.6	
3.5			149.0	39.2	18.5	5.5	2.4	0.8	
4.0				50.2	23.7	7.0	3.1	1.0	
4.5				62.5	29.5	8.7	3.8	1.3	
5.0				75.9	35.8	10.6	4.6	1.6	
5.5				90.6	42.7	12.7	5.5	1.9	
6.0				106.0	50.2	14.9	6.5	2.2	0.6
6.5				123.0	58.2	17.2	7.5	2.6	0.7
7.0				142.0	66.8	19.8	8.6	2.9	0.8
7.5				161.0	75.9	22.5	9.8	3.3	0.9
8.0					85.6	25.3	11.0	3.7	1.0
8.5					95.7	28.3	12.4	4.2	1.1
9.0					106.0	31.5	13.8	4.6	1.2
9.5					118.0	34.8	15.2	5.2	1.4
10					129.0	38.3	16.7	5.7	1.5
12						53.7	23.5	8.0	2.1
14						71.4	31.2	10.6	2.8
16						91.6	40.0	13.5	3.6
18						114.0	49.7	16.9	4.5
20						138.0	60.4	20.5	5.4
25							91.3	31.0	8.2
30							128.0	43.4	11.5
35							170.0	57.8	15.3
40								74.0	19.6
45								92.0	24.4
50								112.0	30.0
55								133.0	35.3
60									41.6
65									48.2
70									55.3
75									62.9
80									70.8
85									74.2
90									88.1
95									97.4
100									107.0

* Head loss in m × 9.806 = head loss in kPa.

Table 16 Friction head loss in water in portable aluminum irrigation pipe with couplers every 6 m (based on C = 130 in Hazen and Williams formula)

Flow rate (L/s)	Nominal pipe size (mm)							
	50	75	100	125	150	175	200	250
	m of water/100 m of pipe*							
0.7	0.6							
0.8	0.7							
0.9	0.9							
1.0	1.1							
1.5	2.4							
2.0	4.0	0.5						
2.5	6.1	0.8						
3.0	8.6	1.1						
3.5	11.4	1.5						
4.0	14.6	1.9						
4.5	18.2	2.3	0.5					
5.0	22.1	2.8	0.7					
5.5	26.3	3.4	0.8					
6.0	31.0	4.0	0.9					
6.5	35.9	4.6	1.1					
7.0	41.2	5.3	1.3					
7.5	46.8	6.0	1.4					
8.0	52.8	6.8	1.6	0.5				
8.5	59.0	7.6	1.8	0.6				
9.0	65.6	8.4	2.0	0.7				
9.5	72.5	9.3	2.2	0.7				
10	79.7	10.3	2.4	0.8				
12	112.0	14.4	3.4	1.1				
14	149.0	19.1	4.5	1.5	0.6			
16		24.5	5.8	1.9	0.8			
18		30.5	7.2	2.4	1.0			
20		37.0	8.8	2.9	1.2	0.6		
25		56.0	13.3	4.4	1.8	0.8		
30		78.4	18.6	6.2	2.5	1.2	0.6	
35		104.0	24.8	8.2	3.4	1.6	0.8	
40		134.0	31.7	10.6	4.3	2.0	1.1	
45		166.0	39.4	13.1	5.4	2.5	1.3	
50			48.0	16.0	6.6	3.1	1.6	0.5
55			57.1	19.0	7.8	3.7	1.9	0.6
60			67.2	22.4	9.2	4.3	2.2	0.8
65			77.9	26.0	10.7	5.0	2.6	0.9
70			89.4	29.8	12.2	5.7	3.0	1.0
75			101.0	33.8	13.9	6.5	3.4	1.1
80			114.0	38.1	15.7	7.3	3.8	1.3
85			128.0	42.6	17.5	8.2	4.3	1.4
90			142.0	47.4	19.5	9.1	4.7	1.6
95			157.0	52.4	21.6	10.1	5.2	1.8
100			173.0	57.6	23.7	11.1	5.8	2.0
110				68.8	28.3	13.2	6.9	2.3
120				80.8	33.2	15.6	8.1	2.7
130				93.7	38.5	18.1	9.4	3.2
140				107.0	44.2	20.7	10.8	3.6
150				122.0	50.2	23.5	12.2	4.1

Table 17 Friction head loss of water in smooth-bore hose
(based on $C = 140$ in Hazen and Williams formula)

Flow rate (L/s)	Nominal inside diameter (mm)										
	16	19	25	32	38	50	64	75	100	125	150
	m of water/100 m of hose										
0.1	2.6	1.1									
0.2	9.3	3.8	0.9								
0.3	19.6	8.1	2.0	0.7							
0.4	33.4	13.8	3.4	1.1							
0.5	50.6	20.8	5.1	1.7	0.7						
0.6	70.9	29.2	7.2	2.4	1.0						
0.7	94.3	38.8	9.6	3.2	1.3						
0.8	121.0	49.7	12.2	4.1	1.7						
0.9	150.0	61.8	15.2	5.1	2.1	0.5					
1.0		75.1	18.5	6.2	2.6	0.6					
1.5		159.0	39.2	13.2	5.4	1.3					
2.0			66.8	22.5	9.3	2.3	0.8				
2.5			101.0	34.0	14.0	3.4	1.2				
3.0			142.0	47.7	19.6	4.8	1.6	0.7			
3.5				63.5	26.1	6.4	2.2	0.9			
4.0				81.3	33.5	8.2	2.8	1.1			
4.5				101.0	41.6	10.2	3.5	1.4			
5				123.0	50.6	12.5	4.2	1.7			
6				172.0	70.9	17.5	5.9	2.4	0.6		
7					94.3	23.2	7.8	3.2	0.8		
8					121.0	29.7	10.0	4.1	1.0		
9					150.0	37.0	12.5	5.1	1.3		
10						45.0	15.2	6.2	1.5	0.5	
12						63.0	21.3	8.7	2.1	0.7	
14						83.9	28.3	11.6	2.9	1.0	
16						107.0	36.2	14.9	3.7	1.2	0.5
18						134.0	45.0	18.5	4.6	1.5	0.6
20						162.0	54.8	22.5	5.5	1.9	0.8
25							82.8	34.1	8.3	2.8	1.2
30							116.0	47.7	11.8	4.0	1.6
35							154.0	63.5	15.6	5.3	2.2
40								81.3	20.0	6.8	2.8
45								101.0	24.9	8.4	3.5
50								123.0	30.3	10.2	4.2
55								147.0	36.1	12.2	5.0
60								172.0	42.4	14.3	5.9
65									49.2	16.6	6.8
70									56.5	19.0	7.8
75									64.2	21.7	8.9
80									72.3	24.4	10.0
85									80.9	27.3	11.2
90									89.9	30.3	12.5
95									99.4	33.5	13.8
100									109.0	36.9	15.2
110									130.0	44.0	18.1
120									153.0	51.7	21.3
130									178.0	59.9	24.7
140										68.7	28.3
150										78.1	32.1

Table 18 Roughness indices for various types of pipe

Pipe material	Roughness factor (ε)
Riveted pipe	3×10^{-3} to 3×10^{-2}
Concrete	1×10^{-3} to 1×10^{-2}
Wood stave	6×10^{-3} to 3×10^{-2}
Cast iron	8.5×10^{-4}
Galvanized iron	5×10^{-4}
Asphalted cast iron	4×10^{-4}
Commercial steel	1.5×10^{-4}
Drawn and plastic tubing	5×10^{-6}

4 PUMPS

Manufacturers specify pumps according to class and type. The three most common classes of pumps are centrifugal, rotary, and reciprocating. These terms apply to the mechanics of fluid movement, not to the pump service. There is a wide variety of pump types, many designed for specific applications.

Fig. 3 classifies the pumps used in agriculture. The miscellaneous class includes pump types usually used for water but not fitting into other classes.

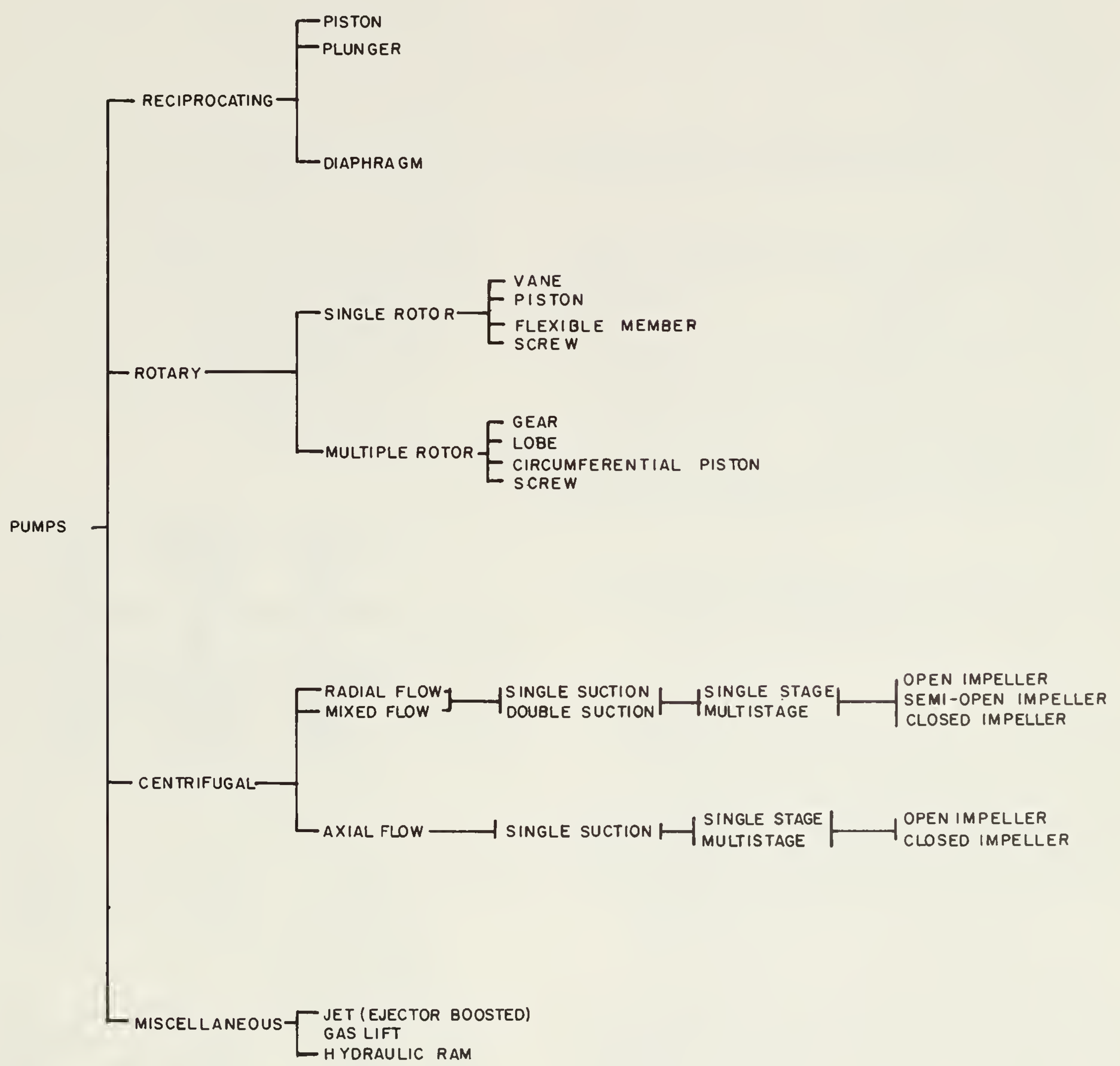


Fig. 3. Classification of pumps.

4.1 General characteristics

The general characteristics of the pump classes vary in:

- flow characteristics
- construction materials
- type of drive

Table 19 lists the various flow characteristics. These characteristics include discharge flow, suction lift, liquids handled, discharge pressure range, capacity, priming, and the effect of increasing head.

The Hydraulic Institute uses the following designations for pump materials:

- bronze-fitted pumps with a cast-iron casing and bronze impeller, casing ring, and shaft sleeves
- all-bronze pumps having all parts made entirely of bronze
- specific-application bronze pumps having the type of bronze specified for particular applications
- all-iron pumps consisting of ferrous metal for the parts of the pump that contact the liquid
- stainless-steel, fitted pumps manufactured from ferrous metal or bronze but with stainless-steel impellers, casing rings, and shaft sleeves
- stainless-steel pumps in which all parts contacting the liquid are made of stainless steel

Pumps serving agricultural applications commonly rely on two types of pump drives:

- direct drive
- belt drive

Direct drive couples the power unit directly to the pump.

Belt drive uses a belt to couple the power unit to the pump. Applications requiring a speed change need belt-driven pumps. Both flat and V-belt drives are used; however, the V-belt drive is most common.

4.2 Centrifugal pumps

Use centrifugal pumps to transport all types of low-viscosity liquids, including abrasive suspensions. Centrifugal pumps can move high-viscosity liquids, but the pump develops low head and capacity and requires increased input power. Water supply and sewage disposal applications, along with farm sprayers, typically rely on centrifugal pumps.

Centrifugal pumps are characterized according to various design parameters, including:

- single- or double-admission intake
- single-stage or multistage operation (using multiple impellers in series)
- relationship between volume and head
- type of vanes, number of blades, and impeller housing

Both single- and double-admission pumps are available as single-stage or multistage. The head at which single-admission pumps operate limits their use. Choose double-admission pumps to lift large amounts of liquid to moderate heights. High-pressure applications require multistage pumps; each pumping stage is used to build pressure. Fig. 4 illustrates flow in single-admission and double-admission, multistage pumps.

Among the variables associated with pump selection, consider impeller design. Several types of impellers have been developed for specific uses. Vane detail, liquid entry to the impeller, and application characterize the various impellers. Impellers typically selected for domestic and agricultural use include:

- open impellers (Fig. 5*a*) with the vanes completely exposed, used in sump dewatering pumps, in which the impellers must move dirty water at low heads, or in laundry pumps

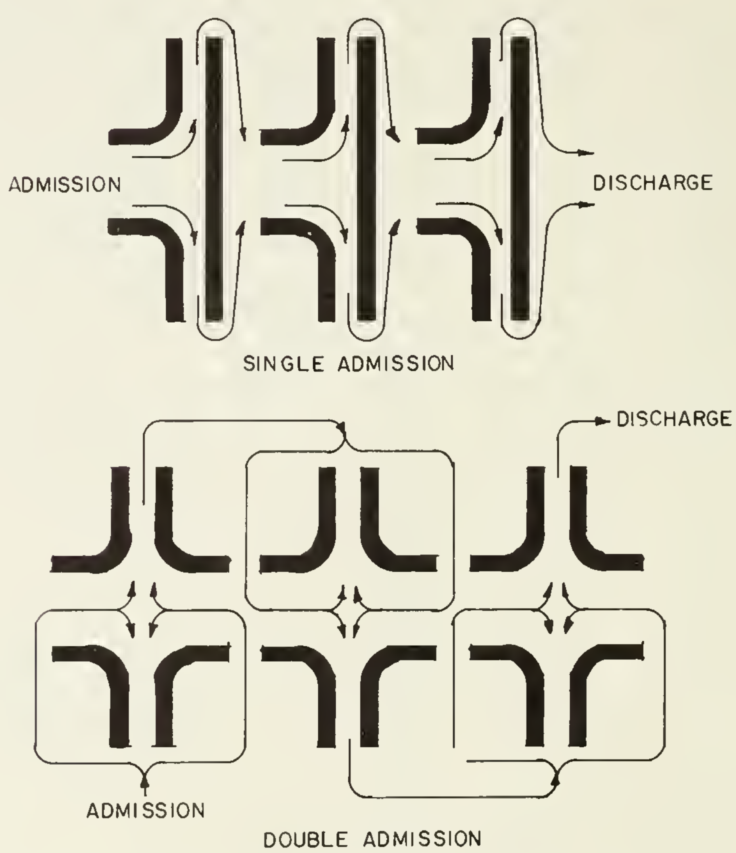


Fig. 4. Single-admission and double-admission multistage centrifugal pumps.

Table 19 Flow characteristics of pumps

	Centrifugal	Rotary	Reciprocating
Discharge flow	steady	steady	pulsating
Maximum suction lift (m)	4.6	6.7	6.7
Liquids handled	• low viscosity • clean • dirty, abrasive liquids with some solids* • slurries with low solid content (diluted manure) • agricultural chemicals	• viscous nonabrasive • agricultural chemicals	• low-to-medium viscosity • clean • dirty liquids with solids • slurries (manure) • agricultural chemicals
Discharge pressure range	low to high	medium	low to highest produced
Capacity range	smallest to largest produced	small to medium	usually small
Effect of increasing head	• decreases capacity • usually increases power, depending on the pump characteristics	• no effect on capacity • increases power	• very small effect on capacity • increases power
Priming	usually required**	not normally required	not normally required

* Impeller may wear excessively when pumping abrasive liquids with closed-impeller pumps.
** Small, self-priming centrifugal pumps are available.

- semi-enclosed impellers (Fig. 5*b*) with a shroud on one side of the vanes, used to handle sewage, liquid manure, and similar types of materials
- closed impellers (Fig. 5*c*) with shroud on both sides of the vanes, used in pumps moving clear liquids at high heads
- propeller and mixed-flow impellers, usually used in high-capacity pumps
- nonclogging impellers, designed to handle solids

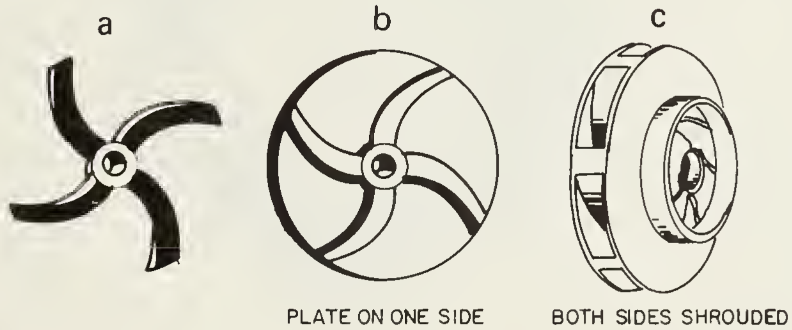


Fig. 5. Impellers. (*a*) Open, (*b*) semi-enclosed, (*c*) closed.

In general, agricultural applications rely on one of four centrifugal pumps:

- volute pumps
- diffuser pumps
- mixed and axial flow pumps
- turbine, or regenerative, pumps

4.3 *Volute pumps* With this type of pump the impeller discharges into a progressively expanding spiral case proportioned so the liquid gradually slows. As a result, most of the kinetic energy converts to static pressure.

Most volute pumps used in the agricultural industry have a single axial inlet and a single radial discharge. Fig. 6 illustrates the volute pump.

4.4 *Diffuser pumps* This type of pump is similar to the volute pump; however, stationary guide vanes surround the impeller. The guide vanes assist in changing the flow direction as liquids pass into the expanding pump case. This arrangement improves the pump's efficiency.

Multistage pumps rely on this principle to redirect liquid flow from one stage to the next.

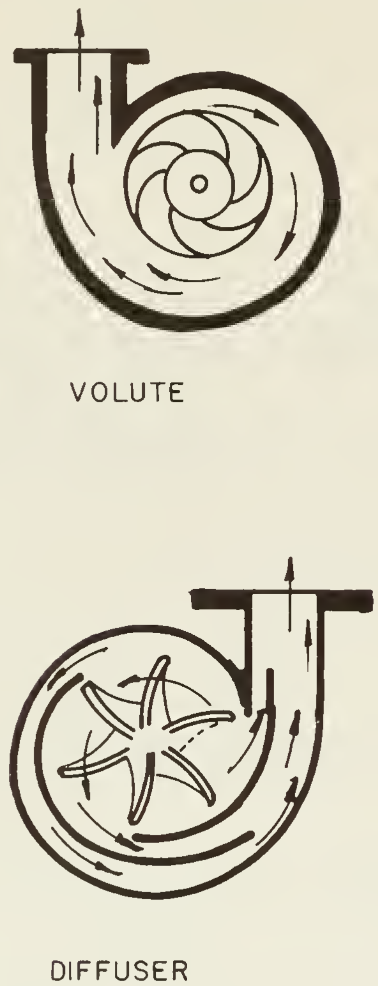


Fig. 6. Volute pumps.

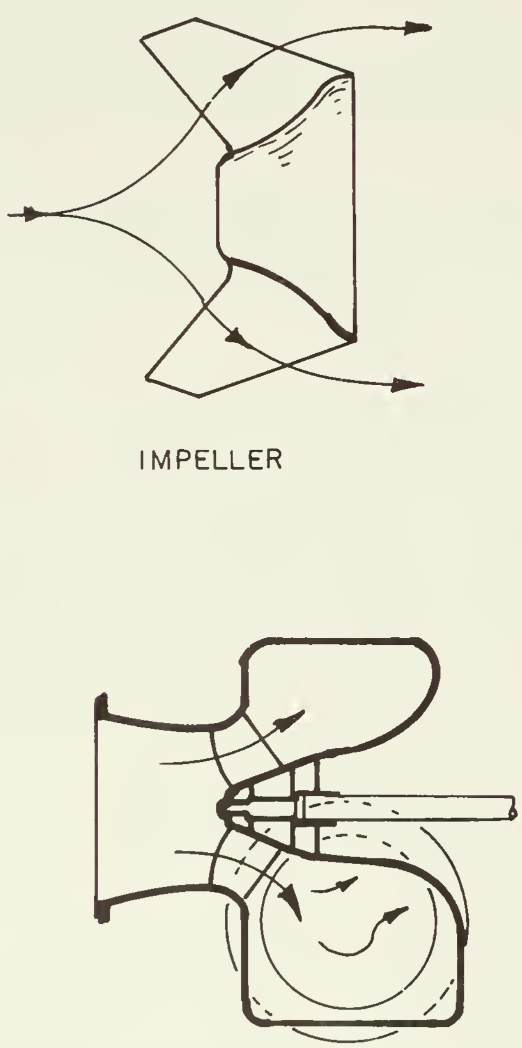


Fig. 7. Mixed-flow pump and impeller.

4.5 *Mixed- and axial-flow pumps* These pumps develop their head by a combination of the centrifugal force and the lift of the rotor vanes on the liquid. Fig. 7 illustrates these two kinds of pumps.

Mixed-flow pumps have an axial inlet and a radial discharge similar to volute pumps. Often high-capacity, low-head pumps use mixed-flow pump impellers to increase operating speed and efficiency, and to reduce pump size.

Axial-flow pumps have an axial inlet and an axial discharge. They operate under zero lift and low head but provide high capacity, as long as the propeller is submerged at all times. Axial flow pumps suit irrigation, drainage, and sewage applications.

4.6 *Turbine (or regenerative) pumps* These are small pumps with multiblade impellers (Fig. 8). The blades are cut into the rim of the impeller and rotate in an annular race chamber. Liquid enters the chamber tangentially to the impeller and moves in and out of the impeller as it flows around the casing. In this way, pressure builds in the pump. After making almost a complete revolution, the liquid exits the impeller and flows into the pump discharge opening.

Use turbine pumps for low-capacity, high-head applications.

4.7 Rotary pumps

Rotary pumps are usually positive-displacement units. They trap liquid at the inlet and carry it around to the discharge nozzle for release.

A rotary pump consists of a fixed casing containing gears, screws, lobes, vanes, or

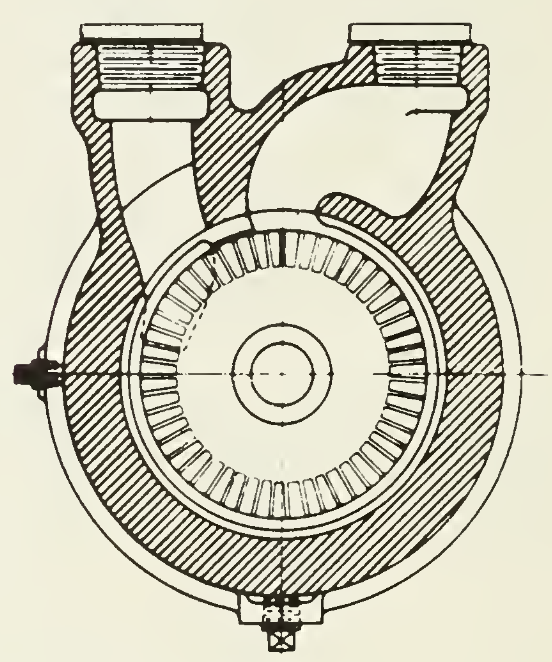


Fig. 8. Turbine, or regenerative, pump.

rollers, depending on the type of pump. The components operate with a minimum of clearance, and discharge flows smoothly.

Rotary pumps usually move viscous liquids; however, they are not confined to this service. Indeed, use rotary pumps for any liquid that is free of abrasives and solids. Typical applications include food and milk handling, hydraulic power transmission, oil feed to burners, and use in sprayers.

Rotary pumps boast several advantages:

- steady flow
- constant discharge, regardless of head
- capacity proportional to speed, for low-viscosity liquids
- self priming

As they do for centrifugal pumps, manufacturers rate rotary pumps for capacity, power input, head, and fluid viscosity. Remember, though, rotary pumps handling viscous liquids such as oil, honey, molasses, or fat operate slower than specifications suggest. Refer to Table 20 for recommended speed reductions.

Six rotary pumps are popular in agriculture:

- cam-and-piston (rotary-plunger) pumps
- screw pumps
- vane pumps
- gear pumps
- lobular pumps
- flexible tube pumps

Table 20 Recommended speed reductions for rotary pumps handling viscous liquids

Liquid viscosity (mm^2/s)	Speed reduction (% of rated pump speed)
130	2
170	6
220	10
320	12
420	14
850	20
1350	30
1700	40
2200	50
4250	55
6300	57
8500	60

4.8 *Cam-and-piston (rotary-plunger) pumps* These pumps consist of an eccentric drive that activates several pistons. Fig. 9 shows a multiplunger pump in which each plunger has a set of inlet and discharge valves. This kind of pump operates at discharge pressures up to 3900 kPa.

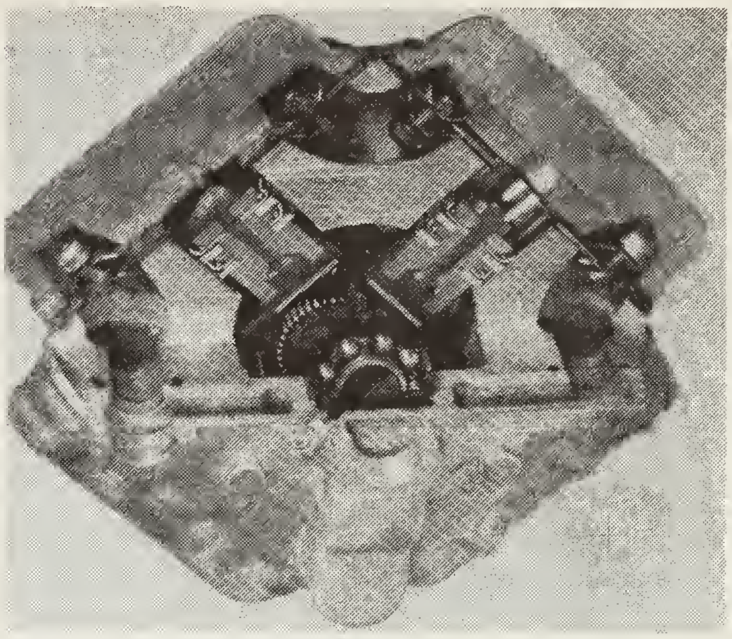


Fig. 9. Cam-and-piston pump.
Source: Seeger-Wanner Corp.

4.9 *Screw pumps* Screw pumps may contain up to three screw-like rotors arranged to turn in an annular-shaped space. The space between the flights of the rotor fills with liquid. As the rotors mesh, this liquid displaces axially. The liquid being moved lubricates the pump. Use screw pumps to handle glue, molasses, paint, asphalt, and chocolate. Fig. 10 illustrates a twin screw pump.

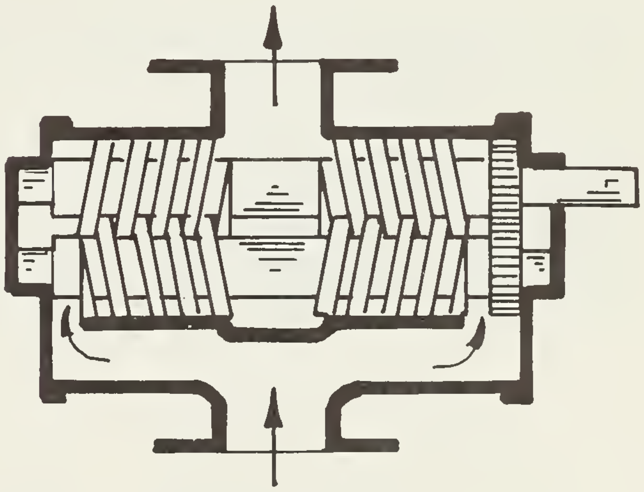


Fig. 10. Twin screw pump.

The Moyno pump, shown in Fig. 11, is a type of screw pump with a flexible rubber stator and a single metal spiral rotor. A helical rotor

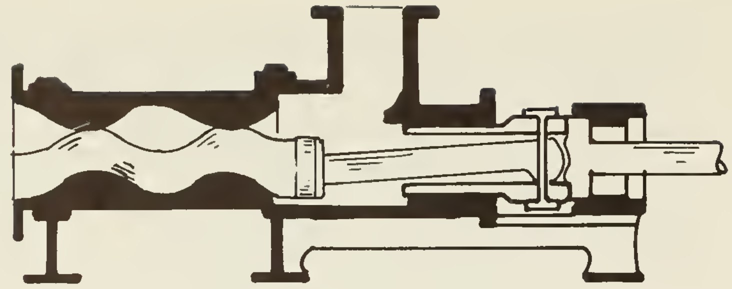


Fig. 11. Moyno screw pump.

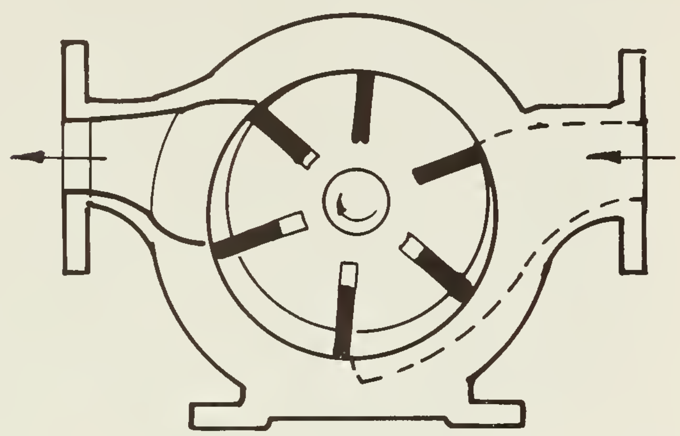


Fig. 12. Sliding-vane pump.

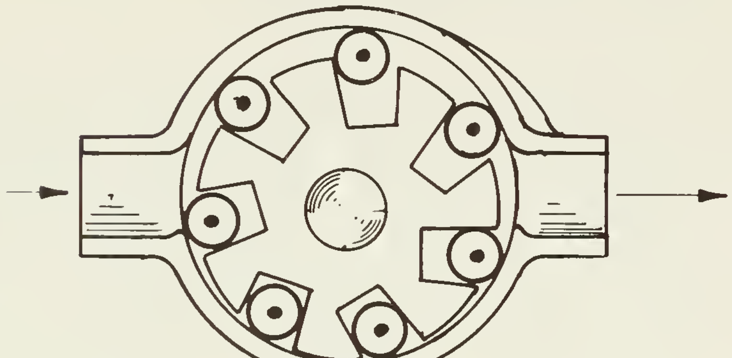


Fig. 13. Vane pump with nylon rollers.

axially displaces liquid. Use Moyno pumps fitted with large-volume irrigation spray nozzles to spread liquid manure.

4.10 *Vane pumps* A vane pump consists of a rotor mounted inside a casing that has been machined eccentrically relative to the rotor axis. Vanes, blades, or rollers fitted to the rotor follow the bore of the casing. Choose from several variations of vane pumps:

- sliding vane pumps
- roller pumps
- flex roller pumps

Sliding vane pumps (Fig. 12) consist of vanes that follow the casing bore. These vanes are usually flat with parallel sides. The vanes are free to slide back and forth and so wear only minimally. Sliding vane pumps operate under a wide range of capacities, speed, and pressures; but they function best at slow speeds. Choose sliding vane pumps to handle mildly corrosive liquids, for example, lubricating and hydraulic oil. Do not use them to pump viscous materials.

Roller pumps (Fig. 13) rely on rubber or nylon rollers in place of vanes. Use this sort of pump to transport agricultural chemicals such as insecticide and herbicide sprays. Roller pumps can handle corrosive liquids; abrasive liquids, however, cause premature wear. Capacities for this kind of pump range from 30 to 260 L/s. They generally operate at pressures of 1000–2000 kPa for speeds of 600–1200 r/min.

The flex rotor pump (Fig. 14) has flexible rubber vanes that rotate inside an eccentric case. Use flex rotor pumps to handle liquids at low pressures: 200–400 kPa. These pumps are self-priming at lifts up to 6 m. They typically handle chemicals, corrosive fluids, detergents, foods, pharmaceuticals, cosmetics, brines, beverages, syrups, and liquid fertilizers.

4.11 *Gear pumps* Gear pumps consist of two or more gears enclosed in a closely fitted casing. As the gears unmesh, liquid fills the space between the teeth. The pump rotation carries the liquid around in the tooth space to where it is displaced when the teeth again mesh. Close tolerance manufacturing assures the efficiency of gear pumps. Shaft speed and the size of the gear cavities set the pump capacity, which can vary from 4 to 400 L/min. Pressures vary from 700 to 20 000 kPa.

Two arrangements of gear pumps are common:

- external gear pumps
- internal gear pumps

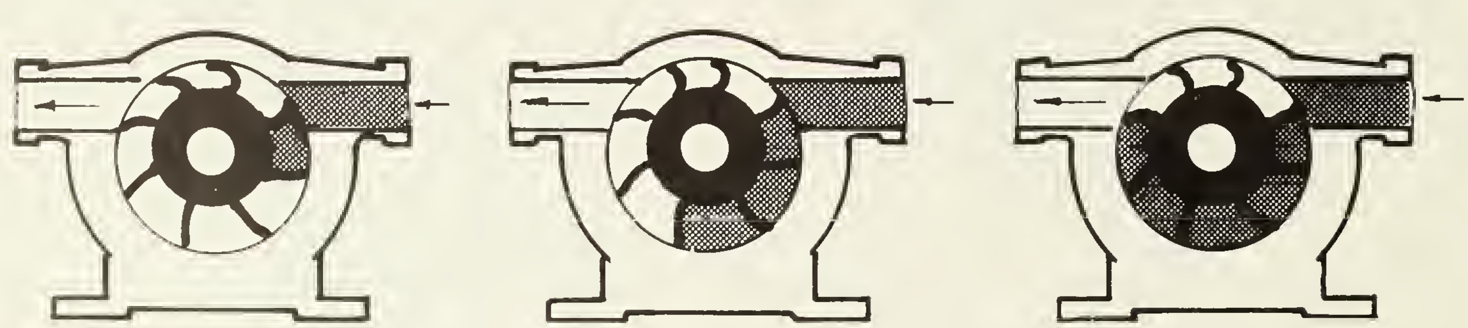


Fig. 14. Vane pump with flexible rotors.

External gear pumps, the most widely used type of gear pump, have all gear teeth cut externally (Fig. 15*a*). Gear design may be spur, helical, or herringbone (Fig. 15*b*).

Choose spur design for low-capacity, high-head applications because the line contact between teeth yields better performance at high pressures. At high speeds, spur gears are noisy and tend to trap liquids at the meshing point, a situation that may cause the shaft to deflect.

Gears designed with a helical shape eliminate trapping problems because the gears engage gradually. However, this design causes a marked side thrust.

Herringbone gears do not produce a side thrust, but they are generally expensive.

At pressures up to 3450 kPa, spur gear pumps transport up to 750 L/min; helical and herringbone designs move up to 1900 L/min. All three types of pumps operate at speeds up to 4000 r/min and can create a suction lift of up to 8 m. Use these gear pumps as transfer pumps or to circulate lubrication oil and refrigerant, to meter chemical feeds, or to add flocculent to water.

Internal gear pumps (Fig. 15*c*) have a single rotor with external-cut teeth enmeshed with an internal-cut gear. A crescent-shaped filler piece between the gears prevents liquid from flowing back to the entry point.

Use internal gear pumps for low-speed, low-pressure applications, to move, for example, viscous materials or liquids sensitive to shear. Pump capacities up to 4200 L/min, pressures up to 1700 kPa, and suction lifts up to 7 m are possible.

4.12 *Lobular pumps* Lobular pumps (Fig. 16) consist of two rotors each having two or three lobes. A system of timing gears drives the rotors to maintain clearance between the lobes. These versatile pumps can handle air and gas-entrained liquids. Lobular pumps can even transport liquids containing large solids without damaging the solids. However, lobular pumps cannot handle abrasives or gritty solids because of the close internal clearances. Typical applications include use in systems pumping soups, fricassees, baby foods, and cookie batter. The capacity of lobular pumps approaches 7500 L/min. Pressures up to 2760 kPa and suction lifts of up to 6 m are common.

4.13 *Flexible tube pumps* The flexible tube, or peristaltic, pump (Fig. 17) depends on the action of rollers at the ends of a rotor to squeeze liquid through a tube. Unique to this kind of pump, the liquid remains isolated from the pump parts, thus limiting contamination. Typical applications include metering chemicals, such as farm fertilizers, and pumping sterile liquids, such as blood plasma. Flexible tube pumps commonly attain pressures of 170 kPa and capacities of 1.5 L/min per tube.

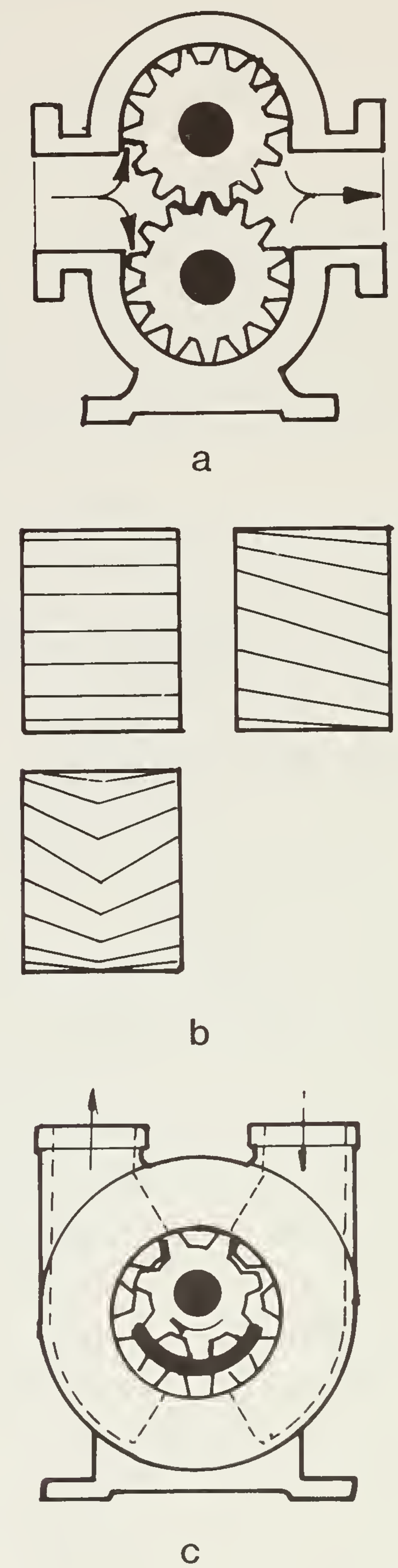


Fig. 15. Gear pumps. (a) External gear pump. (b) Note the various gear teeth arrangements: spur, helical, or herringbone. (c) Internal gear pump.

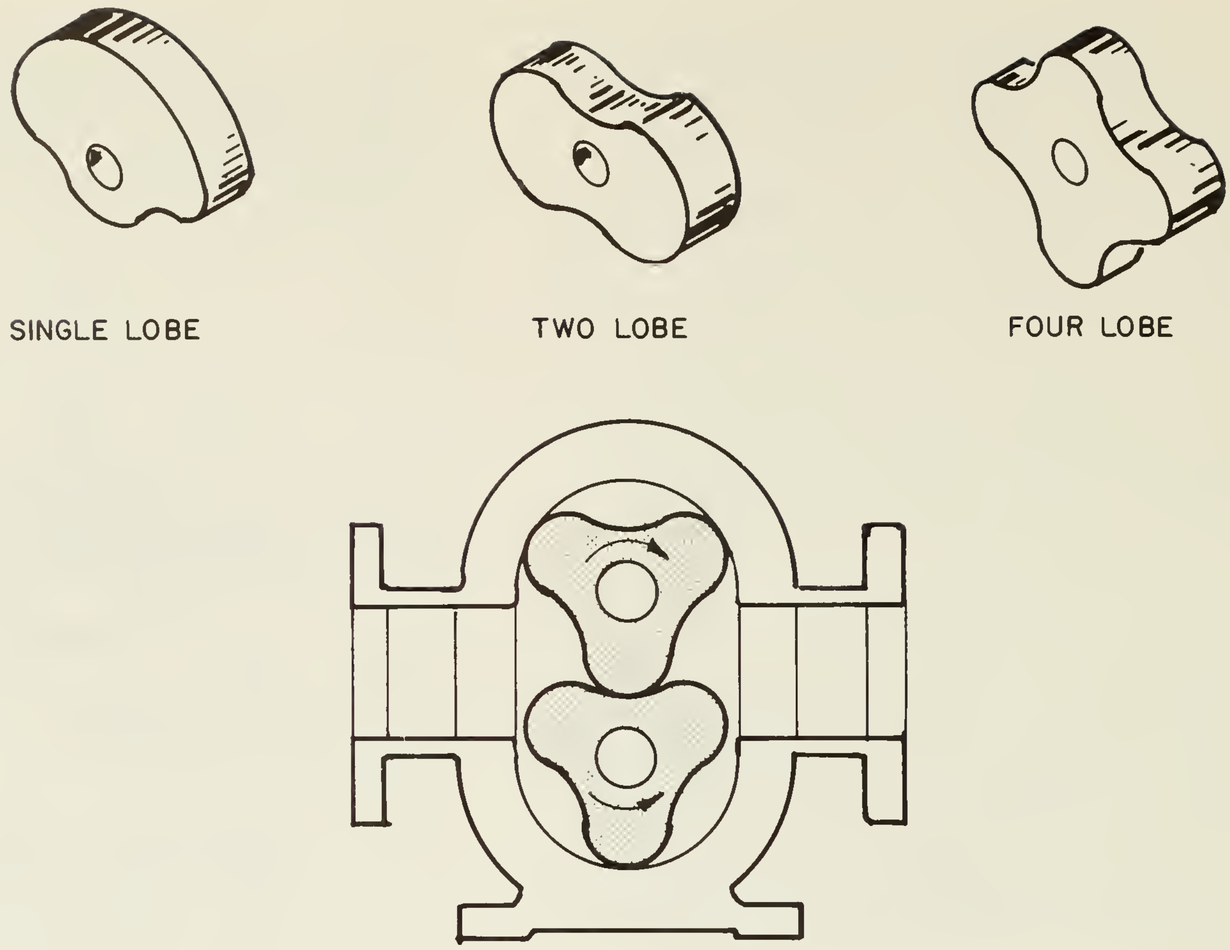


Fig. 16. Lobular pump.

4.14 Reciprocating pumps

Choose from two kinds of reciprocating pumps:

- piston or plunger pumps
- diaphragm pumps

4.15 *Piston or plunger pumps* Piston or plunger pumps are positive displacement systems. Each pump stroke displaces a given volume, regardless of the discharge pressure (Fig. 18).

Piston pumps differ from plunger pumps in two basic characteristics. Firstly, a piston is shorter than the length of the cylinder; a plunger is longer. Secondly, the pump seal in a piston pump is on the piston; plunger pumps have the seals built into the casing.

Use piston pumps for domestic water service and high-pressure sprayers. Plunger pumps frequently serve in deep wells or in systems conveying manure, particularly in dairy operations. Both these pumps can deliver large capacities at high pressures.

4.16 *Diaphragm pumps* Diaphragm pumps handle both clear liquids and those containing high concentrations of solids such as pulps, sewage, sludge, and corrosive materials. A rubber diaphragm replaces the piston and cylinder used in other reciprocating pumps, and the outer edge of the diaphragm bolts to a flange on the pump casing. The flexibility of the diaphragm permits the plunger to move.

Diaphragm pumps are classified as open if the discharge valve is part of the diaphragm, or closed if this valve is part of the pump casing. Fig. 19 shows the construction of both types of diaphragm pumps.

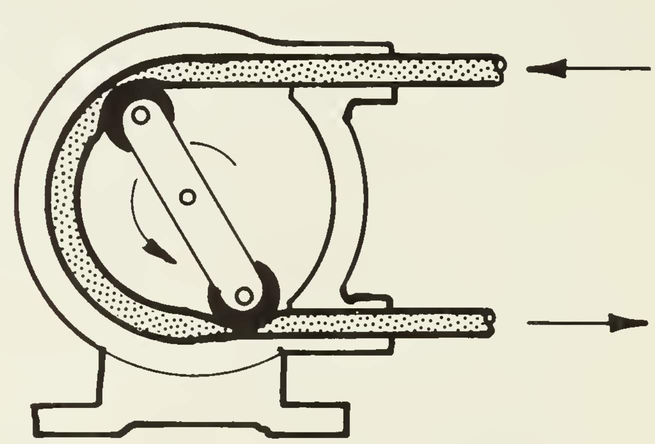


Fig. 17. Pump with flexible tubing.

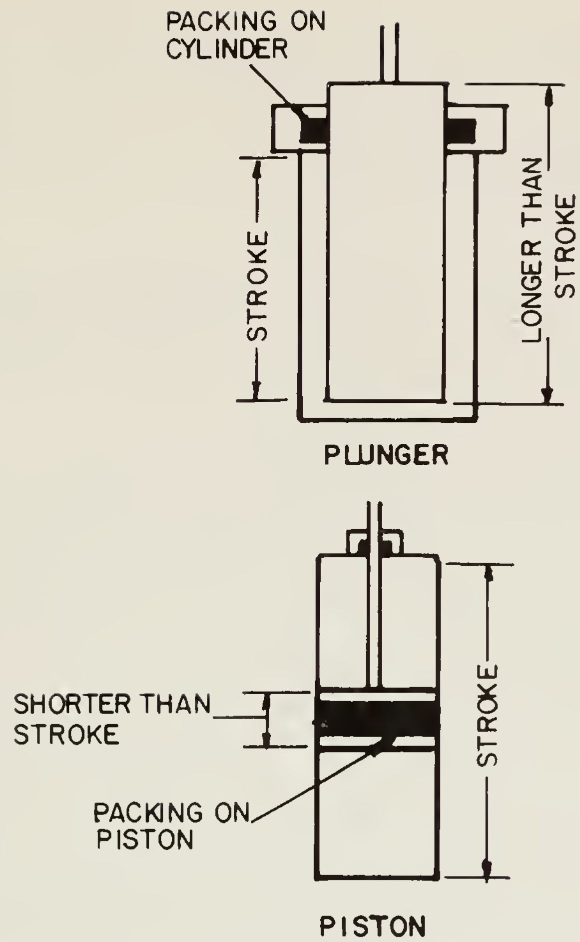


Fig. 18. Comparison of piston and plunger action.

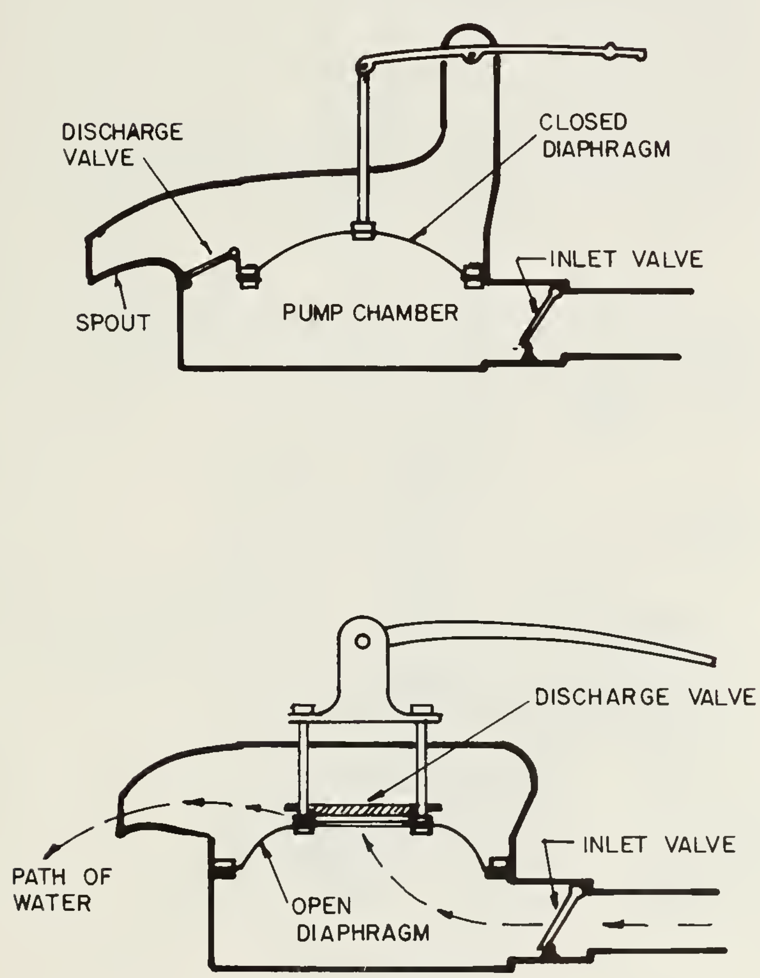


Fig. 19. Diaphragm pump.

4.17 Miscellaneous pumps

A variety of pumps do not fit into the previous classifications. Five are popular in agricultural applications:

- air-lift pumps
- chopper pumps
- jet pumps
- submersible pumps
- hydraulic rams

4.18 *Air-lift pumps* An air-lift pump (Fig. 20) consists of an air compressor and an air line that carries air to a diffuser located near the lower end of a discharge pipe. In operation, compressed air is injected deep into the water inside the discharge pipe. A mixture of air and water forms and rises inside the discharge pipe because the mixture is now less dense than the column of water outside.

Remember two important factors in planning an air-lift pumping system: submergence and the size of the discharge pipe. Measure submergence as the depth of the pump inlet below the pumping water level. Express submergence as a percentage using this equation:

$$\%s_w = \frac{100\, s_w}{s_w + h}$$

where s_w = working submergence (m)
h = lift

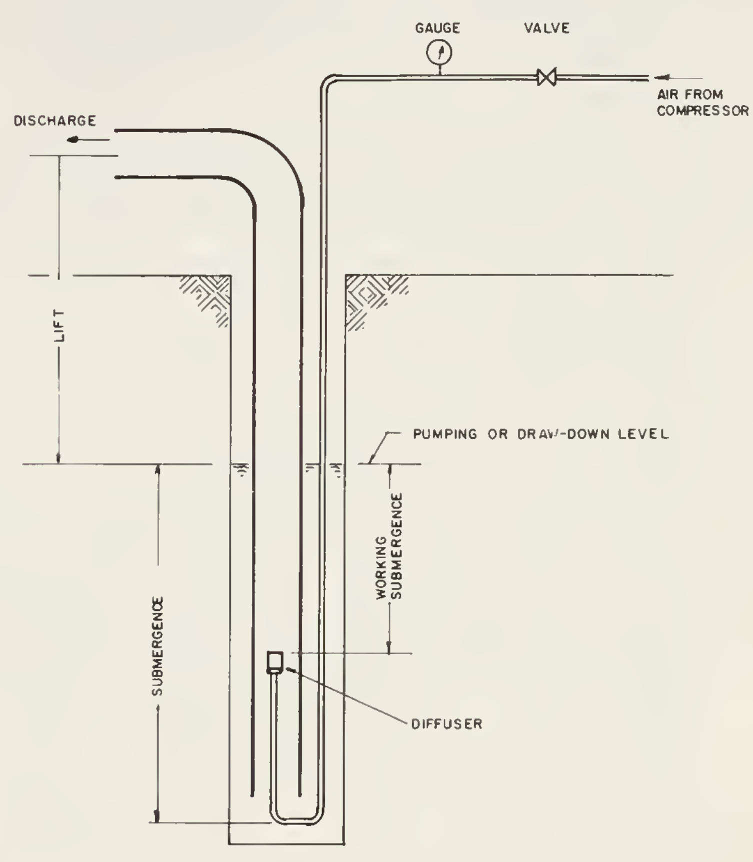


Fig. 20. Air-lift pump.

Table 21 Submergence required for air lift pumping systems

Depth of well (m)	Submergence (%)
up to 15	65
15–30	60
30–60	50
60–100	50

Table 22 Size of air and discharge lines for air-lift pumping systems

Pumping rate (L/s)	Well casing	Nominal pump sizes (mm)	
		Drop pipe	Air line
2–4	90	65	19
4–5	115	75	25
5–6	130	90	32
6–10	150	100	38
10–16	200	130	38

Refer to Table 21 for the minimum submergence values essential for satisfactory operation of the pump.

The cross section of the discharge pipe affects the pump capacity. If the pipe is too large, air does not mix well. If the pipe is too small, excessive friction and inefficient expansion of air bubbles result. Table 22 lists suggested pipe sizes.

Inject air into the discharge pipe through a diffuser or foot piece that has many small holes, approximately 6 mm in diameter. Keep the bubbles small to ensure adequate mixing of air in the liquid.

Set the minimum air pressure to overcome submergence.

$$p = s \times 9.81 \text{ kN/m}^3$$

where p = pressure (kPa)

s = submergence (m)

Increase the pressures to start. This action helps the pump overcome the inertia that commonly exists when the static water level exceeds the pumping level. Table 23 lists air volume requirements and working pressures.

Use air-lift pumps for emergencies or to handle water containing sand or mud. These pumps are, however, inefficient and must discharge into an open tank.

4.19 *Chopper pumps* Chopper pumps resist clogging or jamming, even while handling solids such as plastic, rubber, fiber, and wood. A grinding device at the inlet reduces solids to a slurry of finely divided, easily pumped material. Choose from a variety of chopper pumps that operate with a wide range of capacities and pressures.

4.20 *Jet pumps* Jet pumps combine the centrifugal or turbine action of conventional pumps with an ejector, or jet (Fig. 21). The jet generally sits within the suction lift 6 m from the draw-down or pumping liquid level.

Both shallow-well and deep-well jets are used. For shallow-well jet pumps, locate the jet inside the pump casing or attach it to the pump. Place deep-well jets in the well below the pumping liquid level. Deep-well jet systems are preferable because the pump may be located above ground or offset from the well.

Jet pumps cost less than deep-well turbines or submersible pumps, but are less efficient.

Some of the liquid discharged by the jet pump recirculates through the drive pipe to the jet nozzle. This high-velocity stream creates a low-pressure or suction area that draws liquid in from the well and delivers it into a Venturi tube. The liquid then moves into the enlarged portion of the Venturi tube where the pressure increases. This pressure in turn forces the liquid up the delivery pipe to within the suction range of the pump.

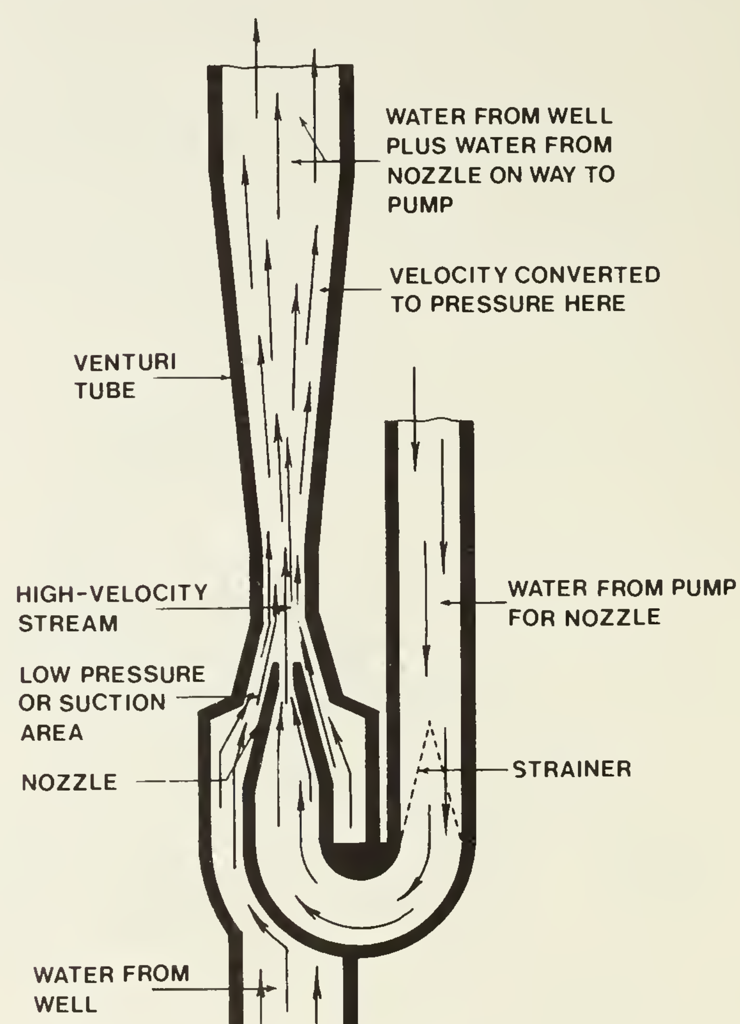


Fig. 21. Jet pump. *Source*: Agriculture Canada.

Table 23 Air volumes and working pressures for air-lift pumping systems

Lift from draw-down level (m)	Submergence (depth of air line below draw-down level) (m)	Air (L per L of water pumped)	Working pressure (kPa)
12	12	3.7	120
	24	2.5	240
18	18	4.5	180
	36	3.0	355
24	24	4.9	240
	48	3.4	475
30	30	5.6	295
	60	4.0	590
42	42	6.4	415
	85	4.7	835
60	60	7.9	590

Jet pumps can operate at depths of 120 m or more; however, for lifts over 46 m some other pump is recommended.

For efficient operation of jet pumps, select the ejector to match the lift. Choose from two types of ejector assemblies. The two-pipe assembly relies on two separate pipes, one to move the liquid and one to supply liquid to the ejector. This two-pipe ejector operates inside well casings over 100 mm in diameter. The single-pipe assembly, designed for smaller wells, uses the well casing as the drive pipe to supply water to the ejector. Adapters allow single-pipe assemblies to fit into most small well casings.

4.21 *Submersible pumps* Submersible pumps consist of a small, multistage centrifugal pump coupled to an electric motor. The pump and motor are built in one compact unit, which can mount on the end of the discharge pipe suspended below the water level. Insulated waterproof wires run down to the motor to supply power to the pump. Submersible pumps are available to fit inside well casings 100 mm and larger. Both impeller and turbine types of submersible pumps are available. They can deliver high pressures and capacities.

Because they operate under water, submersible pumps have several advantages, including high efficiency and exceptionally quiet operation. In addition, priming is not necessary and the cost of frost proofing is usually reduced. However, should the pump fail, the pump and all the piping must be removed from the well. Further, sand or corrosive liquids compromise the operation of submersible pumps.

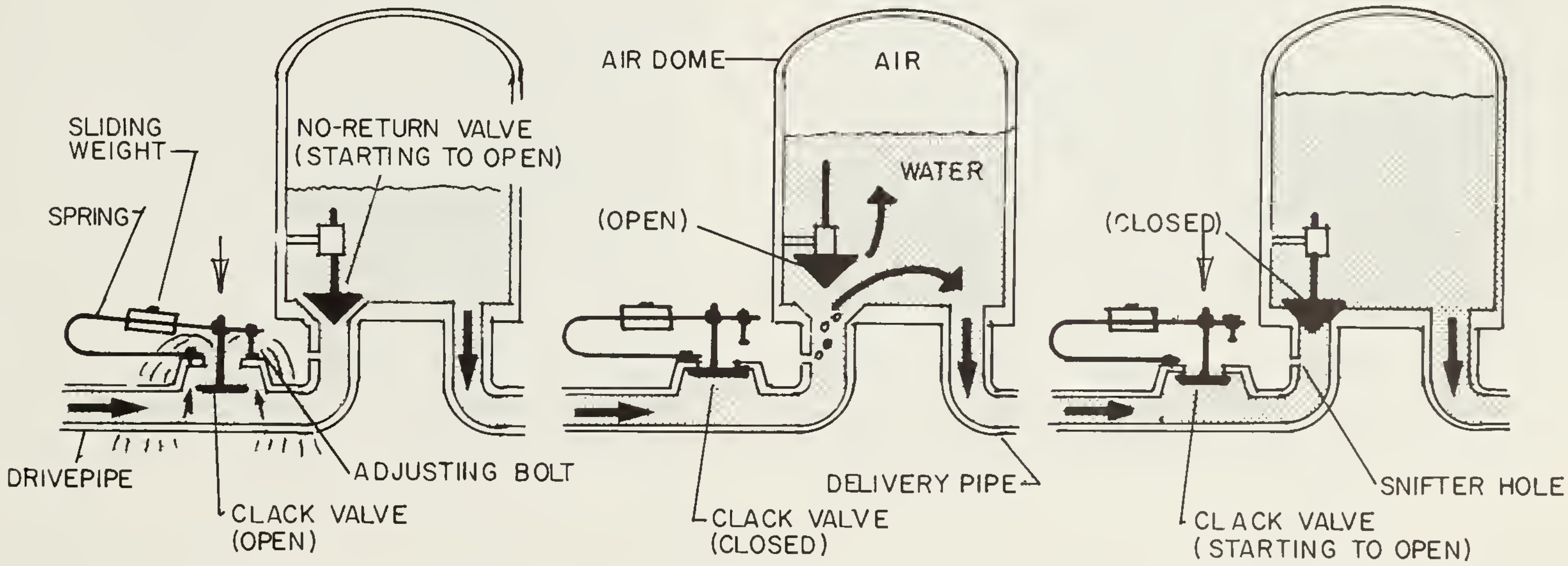


Fig. 22. Hydraulic ram pumping cycle.

4.22 *Hydraulic ram* The hydraulic ram combines a prime mover with a pump. It requires a water supply elevated above the ram. The kinetic energy of the water flowing in the drive pipe from the raised supply drives the ram. This energy produces a pressure in the ram that forces some of the water to a pressure higher than that produced by the static head of the supply alone. Fig. 22 illustrates the pumping cycle of the hydraulic ram.

Hydraulic rams, however, are not very efficient. Consider a ram delivering water at 190 L/min through the drive pipe at 9 m of head. This system elevates the water 15 m at a rate of 20 L/min. The balance of the water is wasted.

Use hydraulic rams to supply water from a stream or spring where no electric power is available.

5 CHARACTERISTICS OF PUMPS AND SYSTEMS

5.1 Introduction

The American Society of Mechanical Engineers (ASME) has developed standard procedures for testing both centrifugal and rotary pumps. Pump manufacturers use these procedures to evaluate pump performance and provide pump specifications. Knowledge of several terms and definitions is important in understanding the standardized data.

Static head is the vertical distance between the free level of the source of supply and the point of free discharge. Alternatively, static head is measured from the source to the level of the free surface of the discharged liquid.

Static suction head refers to the height of the supply liquid surface above the pump centre line.

Static discharge head is the vertical distance between the pump centre line and the point of free delivery of the liquid. Static discharge head is also measured to the level of the free surface of the discharged liquid.

Static suction lift refers to the vertical distance between the liquid supply level and the pump centre line.

Total static head refers to the vertical distance between the supply and discharge levels of the conveyed liquids.

Friction head (measured in metres) is the equivalent head required by the conveyed liquid to overcome resistance to flow in pipes, valves, and other fittings throughout the piping system. Friction head losses occur on both the suction and discharge side of the pump and vary with flow rate of the liquid, pipe size, interior condition of the pipe, and the nature of the pumped liquid.

The resistance of pipe fittings is usually expressed as an equivalent length of straight pipe. Tables 13–17 list friction head losses for water in various types of pipe; Tables 24 and 25 list equivalent lengths of pipe for various fittings. Calculate friction head loss using the Darcy–Weisbach equation. (See section 3.)

Velocity head refers to an equivalent distance through which water must fall to achieve a velocity equal to the average velocity of the water being pumped. Indirectly, velocity head relates to pump energy. Use the following equation to calculate velocity head (H_v):

$$H_v = V^2 / 2g$$

where V = velocity

g = acceleration due to gravity

For most pumping situations, velocity head is small and can be neglected when calculating total head.

Entrance and exit losses occur, due to friction, when the liquid enters or exits the piping system. Except in exceptional situations, these losses are negligible.

Pressure head refers to the pressure at the discharge from a pump as the liquid enters a closed, partially filled tank.

Total suction head is the difference between static suction head and the sum of all friction heads, including any pressure on the liquid on the suction side of the system.

Total suction lift represents the sum of the static suction lift, the friction head, the velocity head, and the inlet losses, minus any pressure on the suction side of the system.

Total discharge head is the sum of the static discharge head, the friction head, the velocity head, and any existing pressure head.

Total pump head measures the energy increase imparted to the liquid by the pump. It is the sum of the total suction lift and the discharge head. Or if a suction head exists, total head is the difference between the discharge and suction heads.

Table 24 Friction head loss due to pipe fittings

Nominal size (mm)	Type of fitting									
	90° elbow	45° elbow	T line flow	T branch flow	gate valve*	globe valve*	check valve*	faucet*	foot valve*	strainer
	Equivalent length of pipe (m)									
6.40	0.70	0.23	0.24	0.73	0.10	6.40	2.19			
9.50	0.94	0.16	0.37	1.07	0.14	6.71	2.22			
12.70	1.10	0.22	0.52	1.28	0.17	6.71	2.44	4.88	1.22	3.05
19.00	1.34	0.28	0.73	1.61	0.20	7.31	2.68	6.40	1.52	3.66
25.40	1.58	0.40	0.97	2.01	0.26	8.84	3.35		1.83	4.27
31.80	2.01	0.52	1.40	2.65	0.33	11.30	3.96		2.13	4.88
38.10	2.25	0.64	1.71	3.02	0.37	12.80	4.57		2.44	5.49
50.80	2.59	0.83	2.34	3.66	0.46	16.50	5.79		2.74	6.10
63.50	2.83	0.97	2.83	3.96	0.52	18.90	6.70		3.05	6.71
76.20	3.35	1.22	3.66	5.18	0.58	24.10	8.23		3.66	7.62
101.60	3.96	1.68	5.80	6.40	0.76	33.50	11.60			

* Fully open condition.

Table 25 Friction head loss caused by insert fittings in plastic pipe

Nominal size (mm)	Type of fitting	
	Insert coupling	Insert adapter
	Equivalent length of pipe (m)	
12.7	0.15	0.30
19.0	0.23	0.46
25.4	0.30	0.61
31.8	0.38	0.84
38.1	0.46	1.07
50.8	0.61	1.37
76.2	0.91	1.98
101.6	1.22	2.74
152.4	1.90	4.27

5.2 System head curves and pump selection

The piping system head versus flow characteristics is a relationship that plays an important role in selecting pumps appropriate to their applications. Analysts use system head curves and pump characteristic curves to find the pumps most suitable for the desired operating range. This selection method applies to centrifugal, rotary, and reciprocating pumps and fans.

For any piping system, the flow rate establishes head loss caused by pipe friction and velocity head. Thus, for any fixed static head condition, the system head increases as the flow rate increases. Static head changes similarly. Add pipe friction to the maximum static head and add velocity head losses to the minimum static head to generate the data for the system head curves.

Establish a pump operating point by superimposing the graph of the pump curve on the system curve. The point at which the two curves intersect defines the operating point of the pump. Then use the operating points to assess the pump's operating range and the suitability of the pump for the application.

The following three examples demonstrate how to develop system head curves.

5.3 *Sample problem 1* Develop a system head curve for the piping arrangement shown in Fig. 23. The pump has a flow rate of 1.5 L/s.

Suction pipe component	Equivalent length (m)
32-mm suction pipe	75.0
one 90° elbow	2.0
one foot valve	2.1
one strainer	4.9
	84.0

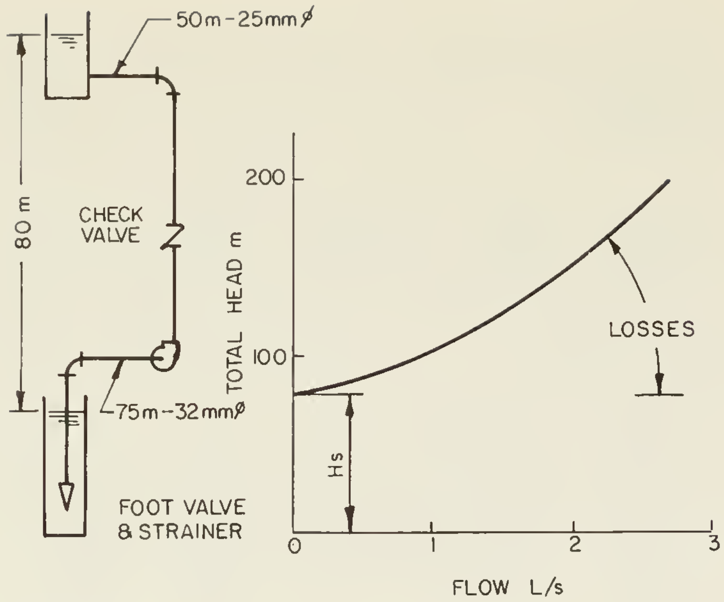


Fig. 23. Pump with a single discharge pipe.

The total equivalent length for the suction pipe (L_s) is 84.0 m.

From Table 13, the friction head loss (H_f) for a 32-mm pipe and a flow rate of 1.5 L/s equals 15.2 m per 100 m of pipe.

$$H_f = L_s \times \frac{H_f}{100}$$

$$= 84.0 \times \frac{15.2}{100}$$

$$= 12.8 \text{ m}$$

where H_f = friction head loss

L_s = suction pipe equivalent length (m)

Discharge pipe component	Equivalent length (m)
25-mm discharge pipe	50.0
one 90° elbow	1.6
one check valve	3.4
	55.0

The total equivalent length for the discharge pipe (L_d) is 55.0 m.

$$H_f = L_d \times \frac{H_f}{100}$$

$$= 55.0 \times \frac{57.9}{100}$$

$$= 31.8 \text{ m}$$

where H_f = friction head loss

L_d = discharge pipe equivalent length (m)

To calculate the total friction head (H_{ft}), the equation becomes:

$$H_{ft} = 12.8 + 31.8$$

$$= 44.6 \text{ m}$$

Use similar calculations to determine total static head and friction head at various other flow rates. The results of these calculations produce data for a table like this:

Flow rate (L/s)	Total static head (m)	Friction head (m)
0.5	80	5.9
1.0	80	21.1
1.5	80	44.6
2.0	80	76.0
2.5	80	114.9

Then use this table to plot a system curve.

Determine the operating point of the system by overlaying a typical pump head capacity curve on the system curve. Select the pump with the most stable pressure and flow specifications and with the highest efficiency for the smallest impeller size or lowest operating speed.

5.4 *Sample problem 2* Develop a system head curve for the pump with two discharge pipes illustrated in Fig. 24.

When there is more than one pipe, plot the system head curves for the pipes independently. Then add the flow rates for the pipes at the same head to obtain the combined system flow.

For this example, assume a flow rate of 2.0 L/s in each discharge pipe.

First, calculate the discharge pipe friction head loss (H_f) for one discharge system.

Discharge pipe component	Equivalent length (m)
32-mm discharge pipe	140.0
one 90° elbow	2.0
one T-line flow	1.4
	143.4

The total equivalent length for this discharge pipe is 143.4 m.

$$H_f = 143.4 \times \frac{25.9}{100}$$

$$= 37.1 \text{ m}$$

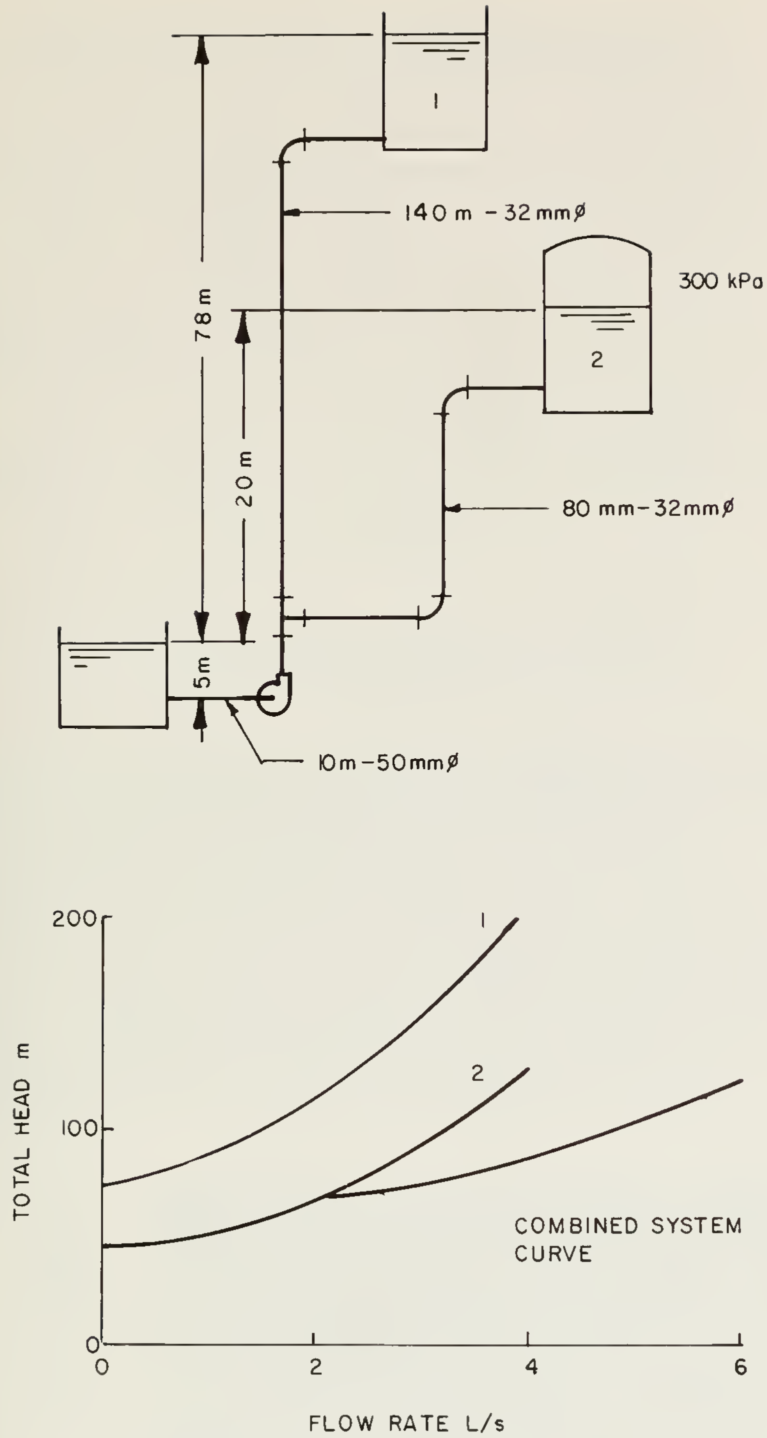


Fig. 24. Pump with two discharge pipes.

Now, calculate the discharge pipe head loss for the second discharge system.

Discharge pipe component	Equivalent length (m)
32-mm discharge pipe	80.0
two 90° elbows	4.0
one T-branch flow	2.7
	86.7

The total equivalent length for this discharge pipe is 86.7 m.

$$H_f = 86.7 \times \frac{25.9}{100}$$
$$= 22.5 \text{ m}$$

In addition, calculate the pressure head (H_p) for this second discharge system. The first discharge system does not have a pressure head because it is an open system.

$$H_p = \frac{p}{\gamma}$$
$$= \frac{300}{9.8}$$
$$= 30.4$$

where H_p = pressure head
p = pressure (kPa)
γ = specific weight (kN/m^3)

Finally, calculate the suction pipe friction head loss for the system. The suction pipe is 10 m long. Note that the flow rate on the suction side is 4 L/s.

$$H_f = 10 \times \frac{13.1}{100}$$
$$= 1.3 \text{ m}$$

Summarizing the data for system 1:

Static suction head	=	–5.0 m
Static discharge head	=	78.0 m
Total friction loss	=	37.1 + 0.7
	=	37.8 m
Total head	=	110.8 m

Summarizing the data for system 2:

Static suction head	=	–5.0 m
Static discharge head	=	20.0 m
Total friction loss	=	22.5 + 0.7
	=	23.2 m
Pressure head	=	30.4 m
Total head	=	68.6 m

Like the previous example, develop a table to list total head data for various flow rates.

Flow rate (L/s)	Total head 1 (m)	Total head 2 (m)
0.0	73.0	45.4
1.0	83.8	52.3
2.0	110.8	68.8
3.0	153.3	94.7
4.0	209.8	129.2

Use a graph of the data in the table to select a pump appropriate to the application. In this example, the minimum flow rate in either system is 2 L/s. This flow rate requires a total head of 110.8 m in system 1, which results in a flow rate in system 2 of approximately 3.5 L/s. Hence, install a pump capable of at least 110 m of head and 5.5 L/s flow rate.

5.5 Pump characteristics

Manufacturers generally use tables or graphs to present various specifications for pumps, most notably power requirements. Table 26 illustrates one method of illustrating pump performance data for a series of small centrifugal pumps. For pumps in this series the maximum lift is limited to about 4.6 m.

Figs. 25 and 26 demonstrate pump performance or characteristic curves. The curves include:

- pump identification
- impeller diameter
- suction and inlet pipe size
- head and capacity
- isoefficiency
- net positive suction head (NPSH)

These sample curves represent the data for a small centrifugal pump having a suction line of 31.75 mm and discharge line of 25.4 mm. Fig. 25 illustrates the head-capacity curves for constant speed and variable impeller diameter. Fig. 26 shows the data for a pump of constant impeller diameter but variable speed.

Note that if the demand on a system changes because of changes in suction or discharge surface levels or in friction characteristics, the total head changes as well.

Among the pump characteristics that influence the systems operation are:

- net positive suction head
- cavitation
- series or parallel installation
- liquids being pumped

5.6 *Net positive suction head (NPSH)* NPSH is the pressure required to force liquid into the pump suction line. All pumps must operate at a minimum NPSH to prevent the impeller blades from being damaged by cavitation. Pump design, capacity, and operating speed determine NPSH.

Not operating at minimum NPSH compromises pump capacity and efficiency. It also causes excessive vibration and noise, reduced life of

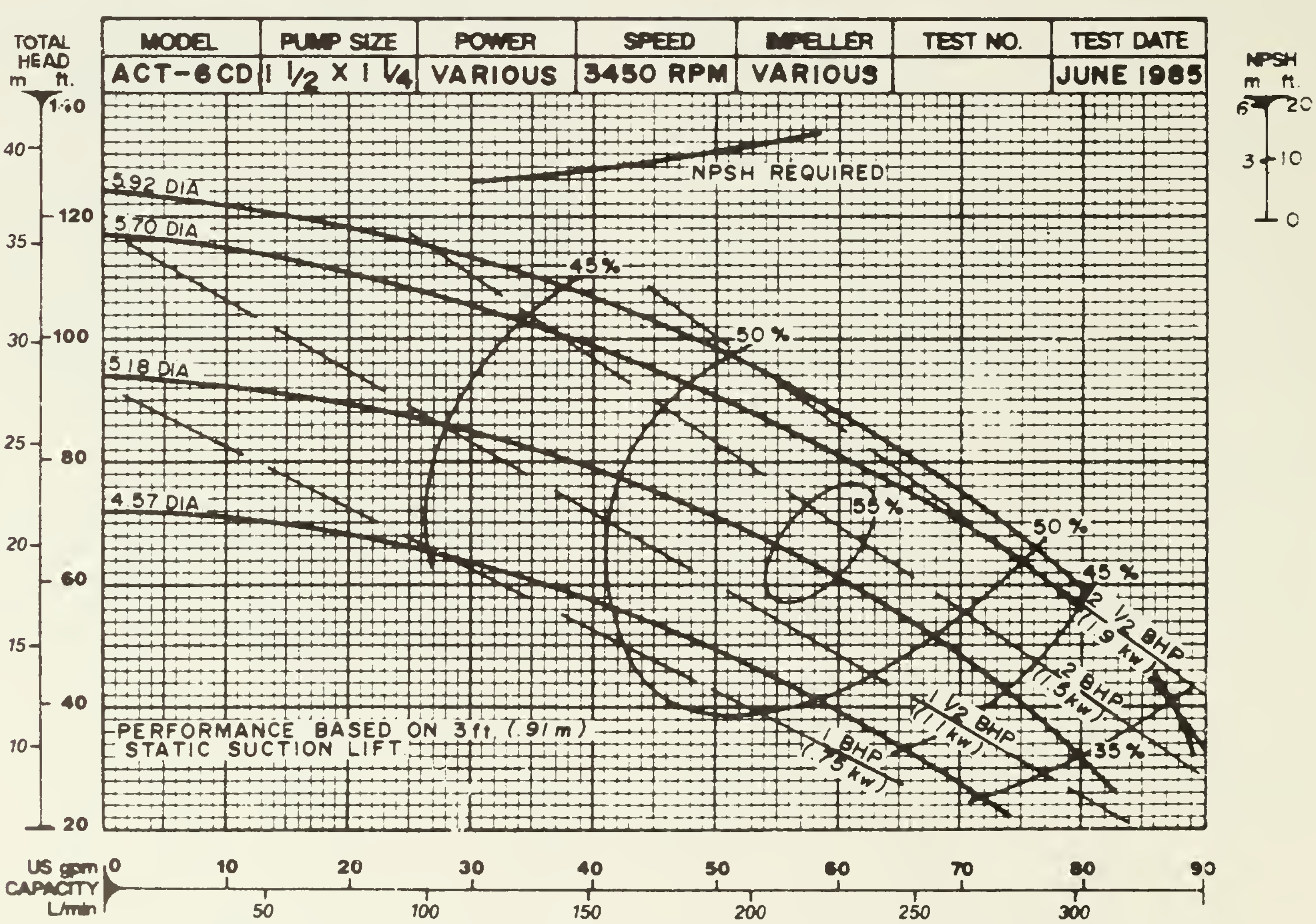


Fig. 25. Performance curves for a centrifugal pump operating at constant speed. These data illustrate how performance varies with impeller diameter.

Table 26 Pump performance data

Motor rating (kW)	Discharge pipe size (mm)	Suction pipe size (mm)	Total head (m)										Shut-off head
			18.3	21.3	24.4	27.4	30.5	33.5	36.6	42.7	48.8	54.9	
			Capacity (L/min)										
2.2	50	75	570	490	415	285							30.5
3.7	50	75		720	700	660	605	550	475	210			44.2
5.9	50	75						755	740	660	530	340	59.4
7.5	50	75								775	740	645	73.1

Notes: Pump operating at 3450 r/min.
Maximum recommended suction lift is 4.6 m.

pump parts caused by cavitation erosion, and damage to the pump from possible vapor lock or from running dry.

NPSH is the total suction head, measured in metres, at the suction nozzle minus the vapor pressure of the liquid, also measured in metres. For rotary and centrifugal pumps, calculate NPSH at maximum flow for maximum line loss. For reciprocating pumps, use both maximum and minimum flows to calculate NPSH and select the pump based on the lower NPSH value.

Depending on the configuration of the pump, use one of the following equations to calculate available NPSH.

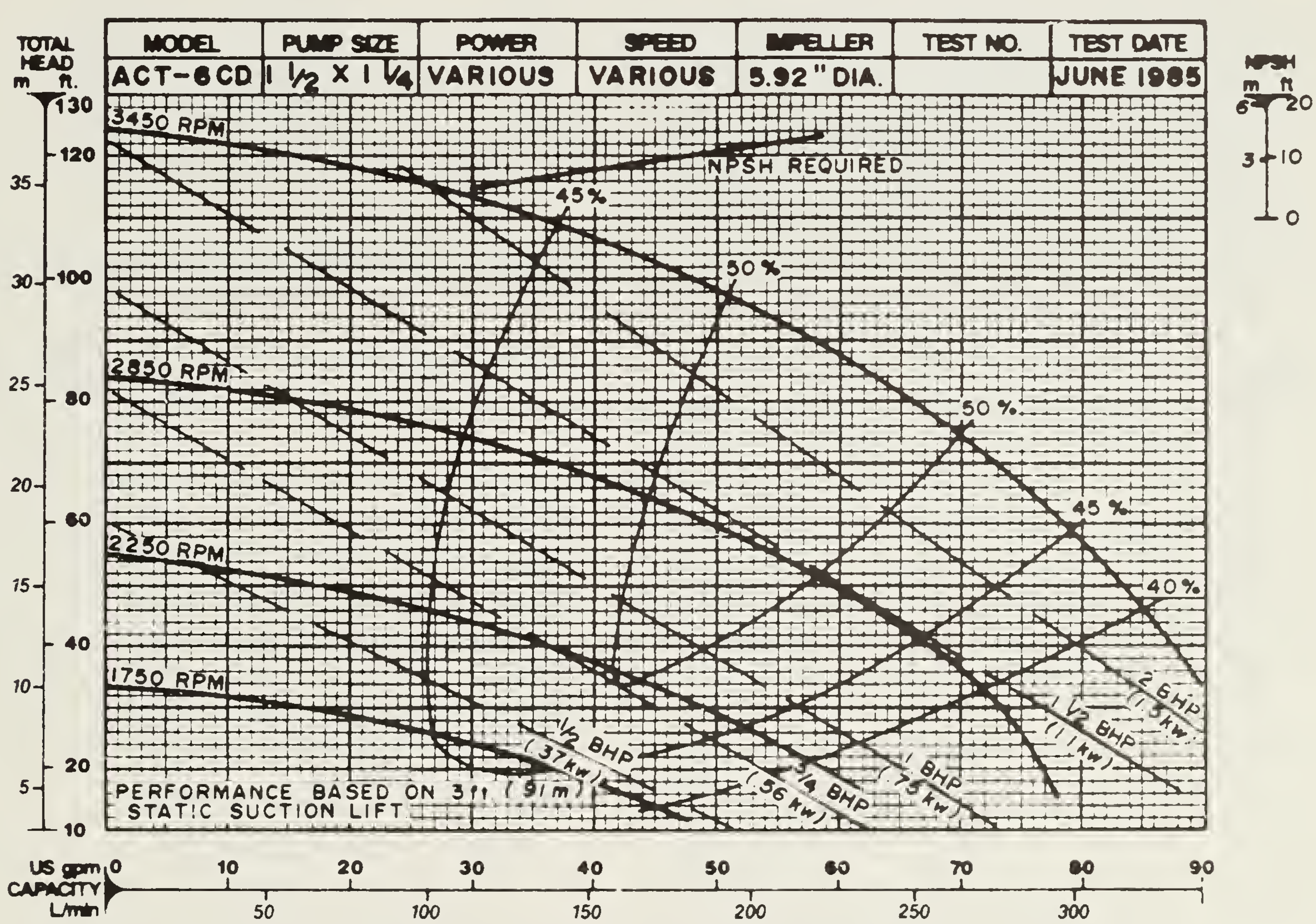


Fig. 26. Performance curves for a centrifugal pump operating at various speeds, but with constant impeller diameter.

When the liquid enters the pump above the centre line and the surface is exposed to atmospheric pressure:

$$\text{NPSH} = (p_a + H_s) - (H_f + p_v)$$

When the liquid level is below the centre line of the pump:

$$\text{NPSH} = p_a - (H_s + H_f + p_v)$$

If a closed tank supplies the liquid to the pump and the liquid enters the pump above the centre line:

$$\text{NPSH} = (p_t + H_s) - (H_f + p_v)$$

If a closed tank supplies the liquid to the pump and the liquid level is below the centre line of the pump:

$$\text{NPSH} = p_t - (H_s + H_f + p_v)$$

For the equations above:

p_a = atmospheric pressure (m)
H_s = static suction head (m)
H_f = friction head loss in the suction line (m)
p_v = vapor pressure of the liquid (m)
p_t = tank pressure (m)

Table 27 lists values for vapor pressure of water. Table 28 lists the atmospheric pressures for various altitudes.

Liquids such as gasoline, refrigerants, and propane, which vaporize readily at normal atmospheric pressures and temperatures, can cause pumping difficulties. These volatile liquids make it difficult to maintain sufficient NPSH. Ensure the available NPSH exceeds that required by the pump; otherwise the liquid vaporizes in the suction line.

5.7 *Cavitation* When pumps operate at very high suction lifts or with high-temperature liquids, there may not be sufficient NPSH to prevent cavitation within the pump from occurring. The cavitation process is complex: bubbles form when the liquid vaporizes at the point of lowest pressure behind the impeller blades. The bubbles then move toward a high pressure area near the blade tips and the bubbles rapidly collapse. As a result, liquid hits the impeller with enough force to gouge metal from the impeller, causing pitting. The collapse of the bubbles also causes the noise and vibration associated with cavitation.

5.8 *Series and parallel installation* For applications with a wide range of demands, operate two or more pumps in series or in parallel. Use pumps arranged in parallel when flow demands are high. Parallel installation allows single-pump operation for periods of low flow demand. To evaluate the performance of the pumping system under varying conditions, plot the system head curve in conjunction with the combined-flow performance curves for the two pumps.

To assess pumps operating in series, plot the composite curve by adding the heads for each pump conveying equivalent capacities. For the pumps operating in parallel, add the capacity of each pump functioning at the same head. To predict the operating point superimpose the system head curve on the composite plot.

Table 27 Vapor pressure of water

Temperature (°C)	Relative density (*d*)	Vapor pressure m	Vapor pressure kPa
10	0.999	0.125	1.23
20	0.998	0.238	2.34
30	0.995	0.433	4.24
40	0.992	0.752	7.38
50	0.988	1.260	12.30
60	0.985	2.030	19.90
70	0.977	3.180	31.20
80	0.971	4.830	47.30
90	0.965	7.150	70.10
100	0.958	10.300	101.30

Table 28 Atmospheric pressure at various altitudes (measured at 15°C)

Altitude above sea level (m)	Atmospheric pressure (kPa)
0	101.3
100	100.1
200	98.9
300	97.7
400	96.6
600	94.3
800	92.0
1000	89.8
2000	79.5
3000	70.1

5.9 *Liquids being pumped* Pump curves expressing head in metres of water are valid regardless of the liquid being pumped. Head expressed in pressure units (kPa) varies for liquids of different relative densities. The following equation describes the relationship between head measured in metres (H) and pressure measured in kilopascals (p).

$$H = \frac{p}{9.81\ \text{kN/m}^3 \times d}$$

where H = head (m)

p = pressure (kPa)

d = relative density

5.10 Pump power requirements

Use the following equation to compute the power required to drive pumps handling incompressible liquids:

$$P = \frac{9.81\ Q\rho H_t}{e}$$

where P = power (W)

Q = flow rate (L/s)

ρ = mass density (kg/m^3)

H_t = total head (m)

e = pump efficiency (obtain this variable from the pump manufacturer)

5.11 Pump capacity

The two most accurate methods for determining pump capacity are:

- to measure the volume of conveyed liquid
- to weigh the amount of liquid delivered in a specific time

However, these methods suit only small amounts of liquid. For larger volumes, use instead Pitot tubes, orifice meters, or mechanical meters. If a weigh tank or other flow-measuring device is not available, use the horizontal discharge method to estimate pump capacity.

5.12 *Pitot tubes* When placed into a flowing stream, a Pitot tube indicates the flow velocity (Fig. 27). This equation determines velocity:

$$V^2 = 2g(H_1 - H_2)$$

where V = flow velocity (m/s)

g = 9.81 m/s^2

H_1 = piezometric head at point 1

H_2 = piezometric head at point 2

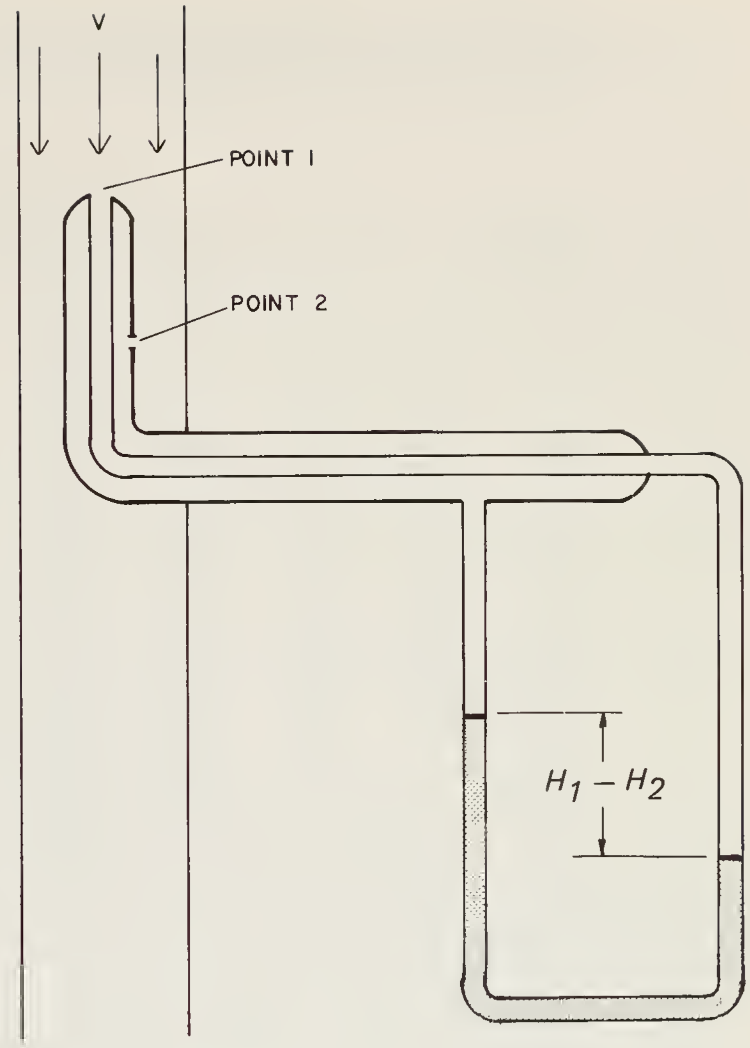


Fig. 27. Pitot tube.

5.13 *Orifice meters* An orifice meter allows head loss to be measured across a restricted opening in conduit. Determine the flow rate based on the geometric characteristics of the orifice and the properties of the liquid.

The most common orifice meter is the Venturi tube. The hour-glass shape of the Venturi tube streamlines the liquid flow and prevents head loss.

5.14 *Mechanical meters* Mechanical flow meters are generally simpler in design and less expensive than orifice meters. Two sorts are popular:

- turbine flow meter
- rotameter

In the turbine flow meter, a rotor spins in moving liquid at a speed proportional to the velocity of the liquid. The rotameter consists of a vertical tube through which liquid flows upwards. A small element floats in the liquid at a height proportional to the flow rate.

5.15 *Horizontal discharge* To estimate the flow rate from a horizontal pipe, construct an L-shaped measurement instrument as illustrated in Fig. 28. Measure the distance that the top surface of the discharge liquid must travel from the pipe before it drops 100 mm. Then refer to the table in Fig. 28 to estimate the flow rate.

Determination of pump capacity by the horizontal discharge method. To estimate the flow rate from a horizontal pipe construct an L-shaped measuring instrument as shown above. Measure the distance X that the top surface of the water discharging from the pipe must travel before it drops 100 mm. Refer to the table to estimate the flow rate in L/min.

Horiz. Dist. X (mm)	Discharge Rate (L/min) Nominal Pipe Diameter, D (mm)									Average Velocity m/s
	25	32	50	75	100	150	200	250	300	
100	21	37	83	184	316					0.6
125	27	46	104	231	394					0.8
150	32	36	125	276	473	1080				0.9
180	38	65	146	322	553	1260	2190			1.1
205	43	74	166	369	628	1440	2520	4010		1.3
230	48	83	187	416	708	1630	2840	4500	6280	1.4
255	54	93	210	462	783	1800	3140	5030	7000	1.6
280	59	102	229	507	867	1990	3460	5530	8330	1.8
305	64	110	250	553	946	2160	3780	6060	8400	1.9
330	70	119	271	598	1020	2350	4090	6550	9080	2.1
355	76	129	291	644	1100	2540	4390	7040	9800	2.3
380	81	137	312	693	1180	2690	4730	7570	10500	2.4
405	86	148	233	742	1260	2880	5030	8020	11200	2.6
430		157	352	784	1340	3070	5340	8560	11900	2.8
460			375	833	1420	3260	5680	9050	12600	3.0
485			416	878	1490	3440	5980	9546	13200	3.1
510				924	1570	3600	6280	10100	14000	3.2
535				969	1650	3780	6620	10600		3.5
560					1740	3970	6930	11100		3.6
585						4160	7230	11600		3.8
610						4310	7570	12100		4.0

For other than standard diameter pipes the flow may be estimated by using the following formula:

Q (L/min) $= 3.37 \times 10^{-4} D^2 X$

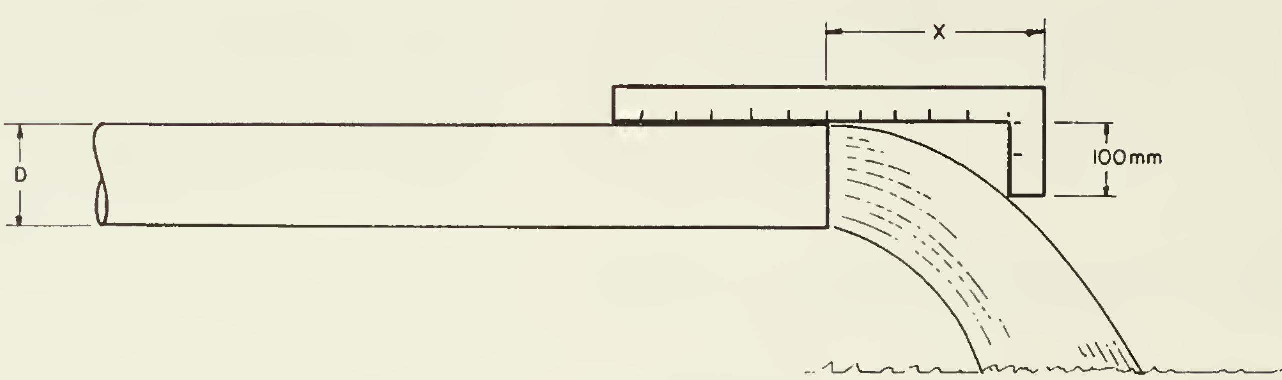


Fig. 28. Horizontal discharge method of determining pump capacity.

5.16 Water hammering

Water hammering occurs in closed pipe systems when the velocity of the liquid changes suddenly. Starting or stopping a pump or suddenly opening or closing a flow-control device, such as a valve, can initiate hammering. The unpleasant noise of water hammering results from a pressure wave as the liquid velocity changes rapidly. A very large pressure rise may cause extensive damage to pumps, piping, and fittings.

Estimate the magnitude of the pressure rise with this equation:

$$p = V \cdot V_r / g$$

where p = pressure rise (m)

V = velocity of the pressure wave in the pipe (m/s)

$= 1480/(1 + K_E \cdot R)^{0.5}$

V_r = change in liquid velocity (m/s)

g = 9.81 m/s^2

K_E = elastic modulus of water/elastic modulus of the pipe

R = ratio of pipe diameter to wall thickness

Here are some values of K_E for common pipe materials:

- steel 0.010
- wrought iron 0.0107
- concrete/asbestos 0.088
- PVC (rigid) 0.83

For example, determine the pressure rise in 800 m of 100 steel pipe if a valve at the discharge end snaps shut when the liquid velocity is 2.41 m/s and $R = 19$.

$$V = 1480 / (1 + 0.010 \times 19)^{0.5}$$
$$= 1301.7 \text{ m/s}$$
$$p = \frac{1301.7 \times 2.41}{9.81}$$
$$= 320 \text{ m}$$

Use the following equation to determine time required for the pressure wave to travel the length of the pipe:

$$t = L/V$$

where L = pipe length between the pump and the device causing the water hammering (m)

t = time (s)

To solve the problem of water hammering, consider the following:

Lengthen to several seconds the time to stop liquid flow, by installing a flywheel on the pump or drive to extend the pump stopping time. Installing slow control valves also largely eliminates water hammering.

Bleed some liquid from the system. Mount a surge arrestor, water-hammer arrestor, or pressure-relief valve near the pump discharge or near the device causing the water hammering. This arrangement bleeds liquid from the pipe and dampens the pressure wave.

Design the system for low flow rates. Use large diameter pipes.

5.17 Pumping thixotropic and dilatant liquids

Understanding the thixotropic properties of liquids such as grease, syrups, plastics, vegetable oils, shortenings, glues, and varnishes can help in designing pumping systems with smaller pipes. Reducing pipe size increases flow velocities and reduces viscosities to practical values for pumping purposes.

When pumping dilatant liquids such as clay slurries, milk chocolate filled with buttermilk powder, starches, and paints avoid temperatures and concentrations where dilatant properties are most severe. If this is not practical, maintain flow velocities as low as possible by increasing pipe size.

6 PUMPING, PIPING, AND STORING MOLASSES AND FAT

6.1 Molasses

The chemical and physical properties of each type of molasses vary with weather, soil type, refining methods, and the degree of adulteration. Table 29 lists the properties of the various types of molasses.

Table 30 presents the viscosity for an average sample of cane molasses at various temperatures. Because of the wide variation in viscosity that can be expected for commercial molasses, use an average viscosity of 11 000 mm²/s when designing systems to convey molasses. However, if the Brix value is high, select instead a viscosity value up to 55 000 mm²/s.

Molasses does not readily mix at temperatures below 20°C, and pumping at temperatures below 15°C can be difficult. Yet, heating molasses above 45°C may cause caramelization or charring and render the molasses unfit for livestock feed. Hot molasses can also foul the handling system. So never heat molasses above 38°C. Indeed, storing molasses in a room where temperatures are around 20°C obviates heating.

6.2 Storing molasses

Store molasses in steel or concrete tanks. Make steel tanks from material 2.0 mm or 2.8 mm thick. Portable tanks require heavier material 2.8 mm or 3.6 mm thick.

Table 29 Properties of molasses

	Cane	Beet	Hydrol	Citrus
Standard Brix (15.6°C)	78	76–85	75–80	70–73
Total digestible nutrients (%)	67–72	58–61	60	50–54
Total sugars (%)	48–56	45–52	43–64	41–43
Crude protein (%)	2.0–7.5	6–10	1	3–8
Digestible protein (%)	0	4	–	–
Total minerals (%)	8–13	8–12	2–8	4–8
Density (kg/L)	1.41	1.40	1.38	1.35
Specific heat (kJ/(kg·C))	2.09	2.09	2.09	2.09

Table 30 Typical viscosities for 79.5° Brix molasses

Temperature (°C)	Viscosity (mm^2/s)	SSU
15	7920	36 000
20	3960	18 000
25	1980	9 000
30	990	4 500
35	506	2 300
40	253	1 150

Source: B. Maddox.

Since the pH of molasses exceeds 5.5, it is not corrosive to steel. Water vapor collecting on the inside of steel tanks can, however, cause rusting. Control this potential problem by ventilating the tank with two pipes 75 mm or larger in diameter.

If choosing concrete storage tanks, construct them of monolithic reinforced steel. Seal the inside with a plastic liner or a reliable concrete sealer.

Fit storage tanks with top openings for easy gaging and cleaning. Design the bottom of the tanks to slope 4% to the discharge pipe or sump.

For prolonged storage maintain the total sugar content of molasses at 43% or more and ensure the temperature remains below 28°C. If the sugar content falls below 43% and the temperature goes above 23°C fermentation can occur. Fermentation spoils molasses for feeding; it also spoils feed mixed with the molasses.

Some surface fermentation occurs when condensation forming inside the tank runs into the molasses. The fermentation forms a dilute surface film. This film does not damage the rest of the full-strength molasses in the tank. Nonetheless, try to prevent fermentation by ensuring that water cannot leak into a molasses storage tank.

Occasionally air or carbon dioxide is entrained in molasses. Air can enter the material via a leak in the pumping system or by open-air transfer when the molasses is dropped 5–10 m into a storage tank. The sugar extraction process or fermentation can also cause carbon dioxide to be trapped in the molasses. However, pumping problems and errors in volumetric measurement only occur when the molasses entrains more than 20% gas by volume.

6.3 Handling molasses

A conveying system for handling molasses requires special design features.

6.4 *Pumps* Because molasses is so viscous, choose positive displacement pumps, such as gear, vane, or screw types. Centrifugal pumps are not as effective. For low cost and easy maintenance, use bronze or bronze-fitted gear pumps in systems designed for farms.

Run rotary pumps at slow speeds, preferably at one-half to one-quarter the speed recommended for water, to ensure proper filling. Since there is practically no slip when pumping viscous materials, ample pressure develops and the capacity varies with speed.

Locate pumps as close to the molasses supply tank as possible, and preferably below it, so the pump suction remains flooded. Use a large suction pipe from the storage tank to the pump inlet, ideally twice the diameter of the pump inlet. Keep the suction pipe as short and straight as conditions permit.

The power required to pump molasses varies with the discharge quantity and pressures. Table 31 lists average power requirements for rotary pumps. Table 32 compares pump performance for two gear pumps handling molasses at 10°C.

Table 31 Average power requirements for rotary pumps

Pump size (mm)	Power (kW)
25	0.75–12.00
38	1.49– 2.24
50	2.24– 3.73
75	3.73– 5.60
100	7.46–11.19

Table 32 Gear-pump performance: handling molasses at 10°C

	Pump size	
	38 mm	50 mm
Litres pumped per 100 revolutions	30	64
Power (kW) required at 100 r/min and 520 kPa discharge pressure	0.87	1.31
Pump efficiency (%)	30	40

Note: For speeds above 100 r/min increase the power by 0.37 kW for every 50 r/min increase in pump speed for the 38-mm pump and 0.56 kW for the 50-mm pump.

Maintain suction line vacuum at 50 kPa or less. If the vacuum exceeds this limit, allow expansion space for entrained gas.

In systems conveying molasses, install pumps with oversized, spring-loaded pressure relief valves. Set the valve to open when pressures exceed by 10–15% normal operating discharge pressures.

In addition to the relief valve, add a hand-controlled bypass valve on the discharge line. This second valve permits part of the molasses to return to the supply tank or pump inlet when the pump supplies more molasses than required. Besides, operators can open the manual valve before starting the pump motor and thus reduce discharge pressure, especially for cold molasses.

6.5 *Piping and valves* In molasses-handling systems undersized pipes cause significant performance problems. Table 33 lists the recommended pipe sizes for the suction and discharge pipes of a molasses-handling system.

In general, use black iron pipe for handling molasses. If the pipe is exposed to weather, choose galvanized pipe. Alternatively, use plastic pipe, provided it can withstand the pressure at the operating temperature. Install pipe lines as straight as possible. Where bends are required, use two 45° elbows rather than one 90° elbow.

Use the following equation to estimate the pressure loss for laminar flow in clean smooth pipe:

$$p = \frac{4.08 \times 10^3 \mu Q}{D_i^4}$$

where μ = viscosity (Pa·s)

Q = flow rate (L/s)

D_i = inside pipe diameter (mm)

p = pressure loss in pipe (kPa/100 m of pipe)

6.6 *Metering* Incorporate flow meters into a pumping system to monitor both the amount of molasses mixing into feed and the total amount handled. Because gas entrainment or changes in molasses viscosity may cause metering errors, design the piping system so operators can draw off some molasses and check the meter calibration. For farm applications use a weigh tank with a batch-mixing process to do calibrations. Operators weigh molasses on a scale and then pump the material into a mixer, or allow it to flow there by gravity.

Meters indicating flow rate are also available. These usually consist of a generator activated by the flowing molasses. A meter calibrated in some convenient flow-rate units indicates the output from the generator.

Table 33 Recommended sizes for suction and discharge pipe handling undiluted molasses at 10°C with a maximum suction of 68 kPa and a maximum discharge pressure of 500 kPa

Pipe length (m)	Flow rate (L/m) 0–7.5	7.5–20	20–40	40–80
	Pipe size (mm)			
Suction pipe				
0–1.2	50	64	75	100
1.2–2.5	64	75	100	125
2.5–6.0	75	100	125	150
Discharge pipe				
0– 1.5	32	38	50	64
1.5– 3.0	38	50	64	73
3.0– 7.5	50	64	75	100
7.5–15	64	75	100	125

6.7 Tallow

The unique characteristics of tallow — or fat — demand a handling system with special features.

6.8 *Storage* Use vertical or horizontal tanks made of steel or concrete to store fat. If choosing concrete tanks, however, be sure to treat the interior surfaces with a nonsoluble, nontoxic coating. Warm fat penetrates and softens the walls of porous, untreated concrete. Apply the coatings to the clean concrete walls of the tank before any fat enters it.

Equip all tanks that store fat with a weatherproof top vent made from pipe 75 mm in diameter or larger. Empty the tanks periodically for cleaning. To facilitate cleaning, two small tanks are preferable to a single large one.

6.9 *Handling* Pipes, valves, and fittings for systems handling fat should be made of iron or steel. Fat handled in copper or brass equipment becomes rancid quickly.

6.10 *Heating* Avoid overheating fat or introducing water into it. Store the fat as cool as possible, yet keep it fluid enough to handle. Where fat is being used daily, store it at 50°C and preheat it to 60–90°C before mixing the fat with feed.

Use steam coils to heat fat in storage. Install steam coils near the bottom of the storage tank. Introduce the steam lines at the top of the tank and direct the lines down one side to the coils. This configuration permits the hot steam to melt a vertical channel through the stored fat, which allows better circulation of the heat and faster heating. It also prevents high pressures developing under a layer of unmelted fat.

Keep steam lines tight to prevent the entry of moisture into the fat. If allowed to hydrate, fat forms a sludge that can corrode conveying equipment.

6.11 *Pumping* Use positive-displacement pumps for handling fat. (See Section 4.7 for more information on positive-displacement pumps.) Choose a pump made of iron or stainless steel. Install iron gate valves or plug cocks to control flow. If throttling is necessary, however, use globe valves.

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